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I.

PROGRESSIVE TRIALS OF THE STEAM BARGE OF THE COMMANDANT OF THE NEW YORK NAVY YARD.

By Chief Engineer Isherwood, U. S. Navy.

The progressive trials described in the following pages were made during November, 1888, by a Board consisting of Passed Assistant Engineers J. J. Barry and J. H. Perry, and Assistant Engineer W. Strother Smith, United States Navy, with the steam barge appropriated to the use of the Commandant of the New York Navy Yard.

The trials were made in the Hudson river near New York City, for the purpose of ascertaining the efficiency of the machinery, the power required to give the barge its different experimental speeds, and the resistance of the hull at those speeds.

The following are the dimensions of the hull and of the machinery:

HULL.

The hull is of wood, coppered. The rudder is counterbalanced, and there is no rudder post. At the stern a bronze skeg descends 13 inches below the bottom of the keel, passing beneath

the screw, and receiving at its free end the lower pintal or end of the rudder. This skeg was necessary for the protection of the screw, the lowest point of whose periphery had to be below the bottom of the keel in order to obtain a sufficient diameter.

The top of the deck of the barge at the side, amidship, is 18 inches above the surface of the water. The machinery is covered by a house erected on the deck, 7¾ feet wide and 3⅔ feet high above the deck. The flat surface of the end of the house opposed to the air is 28.42 square feet.

The sides of the stem and of the sternpost are chamfered to the form of the hull.

The following are the principal dimensions and proportions of the hull:

Extreme length of deck from forward side of stem to after side of taffrail	55 feet.
Length on water line from the forward side of the stem to the after side of the sternpost	50 feet.
Extreme breadth on deck	10 feet 9 inches.
Breadth on the water line	10 feet 6 inches.
Depth from top of deck beams to the lower edge of the rabbet of the keel, amidship, at side	4 feet 1½ inches.
Depth from water line to lower edge of rabbet of keel	2 feet 7½ inches.
Depth of the keel below the lower edge of its rabbet	5 inches.
Breadth of the keel	4 inches.
Mean draught of water from bottom of keel	3 feet ½ inch.
Angle of dead rise of the greatest immersed transverse section,	13 degrees.
Area of greatest immersed transverse section, exclusive of keel,	18.96 square feet.
Area of water section	368.35 square feet.
Displacement at water line per inch of draught	0.87873 ton.
Displacement to water line, exclusive of keel	17.3875 tons.
Ratio of the length on water line from the forward side of the stem to the after side of the sternpost, to the breadth on the water line	4.76190
Ratio of the greatest immersed transverse section exclusive of keel, to its circumscribing parallelogram	0.68789
Ratio of the water section to its circumscribing parallelogram	0.70162
Ration of the displacement exclusive of keel, to its circumscribing parellopipedon	0.44073
Ratio of the displacement to a prism having the greatest immersed transverse section for base, and the length on the water line for its height	0.64070

Wetted surface of the hull alone	460.65 square feet.
Wetted surface of the keel	60.60 square feet.
Wetted surface of the skeg	2.75 square feet.
Wetted surface of the rudder	9.00 square feet.
Total wetted surface of the vessel	533.00 square feet.

ENGINE.

There is one vertical, direct acting, surface condensing, compound engine, with its two pairs of cranks at right angles to each other. Each cylinder has two piston valves, cutting off the steam by lap, and worked by the Joy valve gear. There are no independent cut-off valves. The Joy valve gear, as used during all the trials, gave the cylinder valves a stroke of $1\frac{5}{16}$ inches, the stroke with full gear being 2.18 inches.

The air-pump and the feed-pump are both vertical, single acting, and worked by the main engine. The surface condenser has a cylindrical shell from end to end.

The injection or refrigerating water is supplied to the condenser by a centrifugal pump 15 inches in diameter and 1 inch in width, driven direct, without gearing, by a separate steam cylinder $2\frac{1}{8}$ inches in diameter and $1\frac{1}{4}$ inches stroke of piston. This cylinder is double acting; the piston rod on one side of its piston is $\frac{7}{16}$ inch in diameter, and on the other side $\frac{13}{32}$ inch in diameter.

The engine was bought ready made from a manufacturer. The following are its principal dimensions and proportions:

Number of cylinders	2.
Diameter of the small cylinder	6 inches.
Diameter of the piston rod of the small cylinder	$1\frac{1}{8}$ inches.
Net area of the piston of the small cylinder exclusive of its rod	27.777324 square inches.
Stroke of the piston of the small cylinder	8 inches.
Space displacement of the piston of the small cylinder per stroke	0.1285987 cubic foot.
Space in clearance and steam passage at one end of small cylinder	0.01081656 cubic foot.
Fraction which the space in clearance and steam passage at one end of small cylinder is of the space displacement of its piston per stroke	0.08333+
Length of the steam port of the small cylinder	$5\frac{1}{4}$ inches.
Breadth of the steam port of the small cylinder	$\frac{5}{8}$ inch.
Diameter of the large cylinder	$10\frac{15}{16}$ inches.

Diameter of the piston rod of the large cylinder	1⅛ inches.
Net area of the piston of the large cylinder exclusive of its rod	87.1270359 square inches.
Stroke of the piston of the large cylinder	8 inches.
Space displacement of the piston of the large cylinder, per stroke	0.4033659 cubic foot.
Space in clearance and steam passage at one end of large cylinder	0.033613 cubic foot.
Fraction which the space in clearance and steam passage at one end of large cylinder is of the space displacement of its piston per stroke	0.08333+
Length of the steam port of the large cylinder	8¾ inches.
Breadth of the steam port of the large cylinder	⅝ inch.
Ratio of the net area of the piston of the large cylinder to the net area of the piston of the small cylinder	3.136624.
Diameter of the crosshead journal	1¾ inches.
Length of the crosshead journal	3 inches.
Diameter of the necks of the connecting rod	1⅛ inches.
Length of the connecting rod between centres of journals	24 inches.
Diameter of the crankpin journals	2¾ inches.
Length of the crankpin journals	3 inches.
Diameter of the crankshaft journals	2¾ inches.
Number of crankshaft journals	3.
Lengths of the crankshaft journals	6¼, 4⅛ and 6¼ inches.
Diameter of the line shaft	2½ inches.
Outside diameter of the thrust ring	5¾ inches.
Area of the thrust ring	20.0277 square inches.
Length of crosshead guide	7 inches.
Breadth of crosshead guide	4 inches.
Diameter of air-pump, (single acting and lifting)	4½ inches.
Diameter of air-pump piston rod	⅞ inch.
Stroke of the piston of the air-pump	3½ inches.
Space displacement of the air-pump piston, per stroke	0.0322136 cubic foot.
Diameter of feed pump plunger	1⅝ inches.
Stroke of feed pump plunger	3½ inches.
Space displacement of feed pump plunger per stroke	0.0042006 cubic foot.
Number of tubes in the surface condenser	91.
Outside diameter of tubes	⅝ inch.
Length of tubes	8 feet.
Aggregate surface of tubes	119.1187 square feet.

BOILER.

There is one vertical boiler of Ward's patent, composed of iron tubes containing the water and surrounded by the gases of combustion.

The shell and the grate are circular. The upper portion of the central pipe is superheating surface.

The grate surface is 7.7 square feet; the heating surface for water, measured on the outside of the tubes, is 293 square feet. Ratio of the heating surface to the grate surface, 38.052. Height of chimney above the level of the grate, 10 feet.

Weight of boiler	2,825 pounds.
Weight of grate bars	264 pounds.
Weight of chimney	146 pounds.
Weight of water in boiler	473 pounds.
Total	3,708 pounds.

Air is supplied to the grate by a Sturtevant blower of monogram No. 2 size, driven by an independent double-acting oscillating steam cylinder of 2⅛ inches diameter and 1¼ inches stroke of piston. This cylinder drives the blower direct—without gearing—and makes at the maximum 2,650 revolutions per minute, producing an air pressure in the ash pit equilibrating a water column 1⅛ inches high.

SCREW.

The barge is propelled by one bronze screw of the modified Griffiths type. The blades—three in number—are arranged equidistant around the axis. The greatest breadth of the blade is at the radius of 6½ inches. The thickness of the blade at the hub is 1 inch, and at the periphery $\frac{3}{16}$ inch.

During all the trials made with this screw, the upper point of the periphery of its disc was 9 inches below the water surface when the barge was stationary.

Diameter of the screw	3 feet
Diameter of the hub	0.50 feet.
Pitch of the screw	4 feet.
Number of blades	3.
Length of the blades in the direction of the axis at the periphery	2 inches.
Length of the blades in the direction of the axis at 6½ inches radius	10¼ inches.
Length of the blades in the direction of the axis at the hub,	0.53 foot.
Fraction used of the pitch at the periphery	0.1250.

Fraction used of the pitch at radius of 6½ inches.....	0 6406.
Fraction used of the pitch at the hub.....	0 3975.
Mean fraction used of the pitch.....	0.4107.
Helicoidal area of the blades.....	3.740454 square feet.
Projected area of the blades on a plane at right angles to axis.....	2.822664 square feet.
Projected area of the blades on a plane parallel to axis.....	2.335500 square feet.
Fraction of a horse-power required for overcoming the surface resistance of the blades of the screw to the water, when the screw makes 60 revolutions per minute,	0.0226153.

THE TRIALS.

Seven progressive trials were made by the Board, the data and results of which, just as observed, are recorded in Table No. 1.*

The respective speeds at which these trials were made, were 4.2705, 5.5843, 6.5336, 6.9142, 7.5790, 8.5247 and 8.8528 geographical miles per hour. The boiler could easily have been made to furnish steam enough for higher speeds, but the engine, with the screw employed and a boiler pressure of 150 pounds per square inch above the atmosphere, could not be made to develope more power.

During the trials, the barge was kept as nearly as possible at 3 feet ½ inch mean draught of water from the bottom of the keel, and was so trimmed that the rabbet of the keel was parallel to the water surface when the hull was stationary. Under these conditions, the upper point of the periphery of the screw was 9 inches below the surface of the water.

The speed was taken by a tested patent taffrail log. The indicators and gauges had all been previously tested. The runs were uniformly three geographical miles, in practically smooth water and light breezes, and on each mile an indicator diagram was taken from each end of each cylinder. The revolutions of the screw shaft were taken by counter. When each set of diagrams was taken, all the corresponding pressures and temperatures were noted, so that for each run of 3 miles there were three complete sets of data, the mean of which in each particular case is given in the following table No. 1.

The intention was to maintain the pressure in the boiler at 150

* Follows page 33.

pounds per square inch above the atmosphere in all the trials, and to regulate the number of the screw's revolutions by the throttle, and this intention was nearly realized.

The number of revolutions made by the circulating pump and by the blower, were ascertained by a mechanical counter pressed by hand against the axes of the respective shafts.

The height of the water columns was directly measured by the difference between the water levels in the two legs of a U tube.

EXPLANATION OF TABLE NO. I.

In the columns of table No. 1 will be found the data and results of the progressive trials, given without correction and just as observed. Each column is lettered at the head for distinction, and contains the quantities belonging to one trial: these quantities are grouped and the lines numbered for facility of reference.

The incomplete column between columns F and G, has been placed in the table because of the determination it affords of the slip of the screw at the experimental speed, and of some other quantities.

Line 1 gives the day and hour at which each trial commenced.

Total Quantities.—Line 2 gives the length of course ran by the barge according to the taffrail log. It was in all cases 3 geographical miles of 6,086 feet. Line 3 gives the time of running the course in minutes and seconds. Line 4 gives the total number of double strokes made by the pistons of the engine, and of revolutions made by the screw, in running the course, as determined by the numbers on the counter at the beginning and end of the course.

Pressures.—Lines 5 and 6 give, respectively, the steam pressures in the boiler and in the receiver, in pounds per square inch above the atmosphere, as denoted by tested spring gauges. Line 7 gives the height of the barometer taken from the standard barometer in the Navy Yard. Line 8 gives the "vacuum in the condenser", or more properly the pressure in the condenser below the atmospheric pressure, in inches of mercury, as denoted by a tested gauge. Line 9 gives the fraction used of the throttle valve opening.

Temperatures.—The temperatures on lines 10, 11, 12 and 13 were taken by ordinary mercurial thermometers graduated according to the Fahrenheit scale. Line 10 gives the atmospheric temperature. Lines 11, 12 and 13 give, respectively, the temperatures of the injection or river water entering the surface condenser to condense the exhaust steam; of the same water when discharged from the condenser after being driven through it by the centrifugal pump; and of the feed water entering the boiler. It will be observed that in trials D, E, F and G, the temperature of the feed water (line 13) is less than that of the discharge water (line 12), and this was due to the fact that the capacity of the feed pump worked by the engine being too small to feed the boiler, river water had to be supplemented by the donkey pump.

Auxiliary Engines.—Lines 14 and 15 give, respectively, the number of revolutions made per minute by the circulating pump supplying the injection water to the condenser, and by the blower supplying air to the furnace. In trials A and B the blower was not used, the natural draught of the boiler furnishing sufficient air for the combustion required, but in the remaining trials the blower was used, discharging its air into a closed or an air-tight ashpit, producing the pressure therein given on line 16. The number of revolutions was taken in both cases by a mechanical counter.

Speed.—Line 17 gives the number of double strokes made per minute by the pistons of the engine. These numbers are the quotients of the quantities on line 4 divided by the quantities on line 3. Line 18 gives the speeds of the vessel deduced from the quantities on lines 2 and 3. Lines 19 and 20 give, respectively, the slip of the screw in geographical miles per hour, and in per centum of its speed. The slip of the screw is the difference between its speed expressed by the product of its pitch into the number of revolutions made in a given time, and the speed of the vessel in the same time.

Steam Pressures in Cylinders.—Lines 21 and 23 give, respectively, the mean indicated pressure in pounds per square inch on the pistons of the small and the large cylinders. These pressures include the pressure required to work the engine *per se*, or

unloaded, which being taken at 3.92 pounds per square inch for the piston of the small cylinder, and 1.25 pounds per square inch for the piston of the large cylinder, and deducted from the quantities on lines 21 and 23 respectively, leave the net pressures on the pistons in pounds per square inch as given on lines 22 and 24. To obtain, however, a single expression for the indicated and net pressures producing the horse-power developed by the engine, it is necessary to reduce the pressures on the piston of the small cylinder (lines 21 and 23) to what they would be if applied to the piston of the large cylinder, and to add these reductions to the corresponding pressures (lines 22 and 24) on the piston of the large cylinder. These operations being performed by dividing the pressures on lines 21 and 23 by 3.136624 (the ratio of the area of the piston of the large cylinder to the area of the piston of the small cylinder) and adding the quotients to the pressures on lines 22 and 24, there result the pressures on lines 25 and 26, which are the indicated and net pressures in pounds per square inch that would have had to be exerted on the piston of the large cylinder had the indicated and net horses power been developed by the large cylinder alone.

Horses-Power.—Lines 27 and 28 give the indicated and net horses-power developed by the small cylinder, calculated for the area of its piston in square inches, the speed of piston per minute due to the quantities on line 17, and the pressures on lines 21 and 22.

Lines 29 and 30 give the indicated and net horses-power developed by the large cylinder, calculated for the area of its piston, the speed of piston per minute due to the quantities on line 17, and the pressures on lines 23 and 24.

On line 31 are the aggregate indicated horses-power developed by the engine. They are the sum of the horses-power on lines 27 and 29. On line 32 are the aggregate net horses-power developed by the engine. They are the sum of the horses-power on lines 28 and 30.

Distribution of the Indicated Horses-power.—Lines 33 to 39, both inclusive, show the distribution of the indicated horses-power developed by the engine, in overcoming in the case of the

experimental barge, the various resistances incident to the propulsion of a vessel by a screw.

Line 33 contains the indicated horses-power developed by the engine (the same as on line 31). Line 34 contains the horses-power expended in working the engine *per se* or unloaded; this power is the difference between the indicated and net horses-power developed by the engine; it is what is required to overcome the friction of the moving parts of the unloaded engine, including the line shafting, and in working the various pumps of the engine. If the screw be supposed removed, the powers on line 34 are what are required to work the engine at the different experimental speeds. These powers are directly as the number of double strokes made by the pistons of the engine per minute, the pressure being constant for all speeds of piston for the same engine.

Line 35 gives the net horses-power developed by the engine. This power is what remains of the indicated horses-power developed by the engine (line 31 or 33) after deducting the horses-power on line 34. The net horses-power are the power applied to the crank-pins. They are the commercially available power developed by the engine, or the power external to the engine.

Line 36 gives the horses-power absorbed in overcoming the friction of the load; they are the product of the net horses-power (line 32 or 35) multiplied by the constant co-efficient of friction 0.075: the latter being derived from Morin's experiments to determine the coefficients of friction of well lubricated smooth metallic surfaces moving under pressure. The friction of the load is solely due to, and is proportional to the net horses-power; it has no connection with the friction of the moving parts of the unloaded engine to which it is entirely additional.

Line 37 contains the horses-power expended in overcoming the surface resistance of the screw-blades to the water. These quantities are calculated on the assumptions that the resistance of one square foot of cast metallic helicoidal surface moving in water at the helical speed of 10 feet per second is 0.45 pound, and that this resistance varies in the ratio of the square of the helical speed.

Lines 38 and 39 give, respectively, the horses-power expended in the slip of the screw and in the propulsion of the vessel, obtained in the following manner: The sum of the quantities on lines 36 and 37 being deducted from the quantities on line 35, the remainder is the horses-power expended in the slip of the screw and in the propulsion of the vessel. Now the pressure of the screw-blades producing the recession of the watery fulcrum of the screw, called the slip, produces also the advance of the vessel; consequently the power expended upon these two effects will be in the ratio of the speed of the slip to the speed of the vessel. But when the speed of the screw is represented by 100 the slip of the screw in per centum of its speed (line 20) represents the speed of the slip, and the difference between this per centum and 100 represents the speed of the vessel; hence, multiplying the above remainder due to the deduction of the sum of the quantities on lines 36 and 37 from those on line 35, by the slip of the screw on line 20 considered as a fraction, will give the horses-power expended in that slip (line 38). The quantities on line 38 being deducted from the remainder above described, leave the horses-power expended in the propulsion of the vessel (line 39).

Lines 40 to 43, both inclusive, show the distribution of the net horses-power (line 32 or 35) in per centum of that power. The quantities on lines 40, 41, 42 and 43, are respectively the quantities on lines 36, 37, 38 and 39, expressed in per centum of the quantities on line 32 or 35, and give the power expended on the various effects, relatively; the power applied to the crank pin being taken at 100.

Line 44 gives the thrust of the screw in pounds. This thrust is the pressure produced by the screw blades upon the water: it is what would be shown by a dynamometer applied to the screw shaft when the latter is horizontal, and is calculated as follows: First, ascertain the number of foot-pounds of work in the power applied to the propulsion of the vessel, that is, multiply the horses-power on line 39 by 33,000. Then, determine the speed of the vessel in feet per minute from the quantities on line 18. Last, divide the former by the latter quantities and the quotients

will be the thrusts of the screw in pounds in the different trials. The thrust thus calculated is the thrust in the *horizontal* direction, which is the thrust required to be known. The thrust shown by a dynamometer is the thrust in the direction of the *axis of the screw*, which latter is rarely horizontal.

RESULTS OF THE TRIALS.

The first deduction to be made from the progressive trials is that of the law governing the resistance of the hull in function of its speed.

The following comparison exhibits the different powers of the speed in which the resistance of the vessel increases; the resistance at the different experimental speeds being taken to be represented by the "thrust of the screw in pounds" (line 44 of Table No. 1):

Designation of the trial.	Thrust of the screw in pounds.	Speed of the vessel in geographical miles per hour.	Power of the speed in which the resistance of the vessel increases between successive trials.
A	144.5433	4.2705	
B	296.4816	5.5843	2.6783
C	405.5844	6.5336	2.0255
D	472.1184	6.9142	2.5780
E	599.4837	7.5790	2.6016
F	924.5317	8.5247	3.6842
G	1125.1129	8.8525	5.2038

From the above appears that the resistance of the vessel in function of its speed remained about the same up to the speed of, say, 7.5 geographical miles per hour (trial E); and that between the limits of speed in trials A and E, the resistance of the hull was, as an average, as the 2.4708 power of the speed.

When the speed exceeded 7.5 geographical miles per hour, the resistance increased in a very much higher ratio than the above power of 2.4708. Between the speeds of 7.5790 and 8.5247 geographical miles per hour (trials E and F), the resistance of the vessel increased in the 3.6842 power of the speed; while between

the speeds of 8.5247 and 8.8525 geographical miles per hour (trials F and G) the resistance increased in the still higher ratio of the 5.2038 power of the speed.

In trials F and G, the "squatting" of the hull at the stern became excessive, and was much greater in trial G than in trial F. The great increase of resistance in function of speed in these two trials was undoubtedly due to this "squatting" whereby the form of the immersed solid of the hull relatively to the water surface became enormously changed. The "squatting" is caused by the water not filling with sufficient rapidity the cavity continuously made by the forward movement of the hull, the after body of which descends by its weight until it meets sufficient water to support it. The whole hull is thus inclined more and more to its direction of motion as its speed increases after it once begins to "squat". The principal portion of the water filling the cavity flows into it vertically, due to the fact that the direction is given normal to the maximum pressure, which pressure is at the lowest depth of the cavity or plane of the rabbet of the keel at the stern. The inclination of the hull to the direction of its motion caused as above described, increases its resistance in function of speed; the greater the inclination the greater in some unknown ratio is this increase of resistance. These changes, if carried to the utmost extent, would cause the rabbet of the keel to make an enormous angle with the horizon so that the hull would appear as almost standing on its after end, in which case the resistance would be a maximum. The after body of the hull in its descent, does not fall quite to the point at which it would be wholly supported by the water beneath it, the forward body partly supports the projecting after body, so that the forward body by this additional weight thus thrown upon it becomes itself more deeply immersed and has its resistance correspondingly increased, while the after body has a greater resistance at the same depression than it would have had, had the water met it solidly there. The result is that the resistance of the hull as a whole is greatly increased with increased inclination; the cross area of the cavity made in the water by the moving hull increases by being deepened, due to the "squatting" of the stern, so that the projection of the hull on

a plane at right angles to the movement is greater, and the wetted surface of the hull is greater, as the inclination is greater. Under these circumstances a vessel moving at a velocity which produces "squatting", will ground in a depth of water that will float it when at rest.

The "squatting" of the vessel not only increases its resistance in function of speed, but decreases the propelling efficiency of the screw by augmenting its slip beyond what is due to the resistance of the vessel. This augmentation of slip results from the inclination of the screw's axis to the direction of the vessel's motion, caused by the "squatting" of the latter. If a given distance is passed over by the vessel in an hour, then, supposing its screw to have no slip and its axis to be horizontal, the speed of the screw (pitch $\times$ revolutions per hour) will be the same as that of the vessel; if, now, the axis of the screw be inclined to the horizontal, then, for the same speed of vessel as before, the screw will have to pass in the same time over a distance represented by the secant of the angle of inclination while the vessel passes over a distance represented by radius, and must now have a slip represented by the difference between the secant and the radius of the angle. Whatever slip the screw may have, under any conditions, an increased inclination of its axis will increase that slip in the manner described.

The increased slip of the screw due to increased inclination of its axis, may be further shown by forming a right angled triangle, the base of which is represented by the speed of the vessel in any given time, the hypotheneuse by the axis of the screw at the given angle of inclination, and the height by the vertical opposite the angle and cutting the hypotheneuse. If the "thrust of the screw" in the direction of its axis be represented by the hypotheneuse, the portion of that thrust expended in propelling the vessel will be represented by the base of the triangle (supposed horizontal), while the portion represented by the height of the triangle will be wholly expended upon the water in the vertical direction producing only displacement of that water, or slip to the full value of the vertical pressure. Of course, the vertical component of the thrust tends to lift the stern of the vessel, but if

the weight of the stern is greater than the vertical component, and it almost always greatly exceeds it, this tendency will not produce any sensible result.

Furthermore, every increase in the slip of a given screw, is attended by an increase in the power required to be expended in overcoming the resistance of the surface of its blades to the water, because the greater the slip the oftener must the screw be revolved for a given speed of vessel, and the power expended in overcoming the resistance of the surface of the blades to the water is in the ratio of the cube of the number of revolutions made by the screw in a given time.

Hence, the increased inclination of the axis of the screw of a given vessel, caused by the "squatting" of the latter, increases all the losses of useful effect by the screw in a higher ratio than is due to the increased resistance of the vessel in function of speed. Both the power expended in the slip and in overcoming the surface resistance of the blades are increased by the "squatting" beyond what is due to the same resistance of vessel without "squatting".

A vessel, therefore, can only be forced above the speed at which "squatting" commences by a disproportionate expenditure of engine power, the ratio of which disproportion rapidly increases as the "squatting" becomes more and more. This is why, at high speeds of vessel, the engine power required increases so enormously above what is due to the theoretical law of the cubes of the speeds. The change in the form of the immersed solid of the hull, relatively to the water surface, and the exaggeration of the losses of useful effect by the screw, renders impossible all calculation of the performance of a vessel beyond the speed at which squatting commences. Experimental knowledge in this case is the only guide.

From the foregoing there evidently follows that the axis of the screw shaft, for the maximum effect from a given screw, should be placed in such a position in the vessel that it will be horizontal when the vessel is at the intended speed. Any deviation from this position involves a loss of useful effect by the screw, unnecessary, and large in proportion as the angle of inclination of the axis to the horizontal is large.

Further, the "squatting" of the vessel causes an increase in the slip of the screw above what is due to the vessel's resistance in function of speed, and above what has been above described, namely, the increase of slip due to the failure of the ascending or closing in water at the stern of the vessel to get "solidly" upon the surface of the blades. As long as the vessel does not "squat", this water necessarily reaches the entire blade surface, but when the "squatting" commences such is no longer the case, for the "squatting" itself being an effect of, is a proof of, deficiency of water at the stern; hence, only a portion of the blade surface of the screw acts propulsively on the water after "squatting" commences, and this portion becomes less and less with greater and greater "squatting". Probably the largest portion of the increased slip of the screw, as observed, is due to the failure of the water to reach the entire blade surface after the vessel passes the limit of speed at which the "squatting" begins. This opinion is confirmed by the universal fact that, owing to difference of pressure upon the screw surface in the lower half of the screw disc and in the upper half, a screw vessel with its helm amidship always has a tendency to turn from a straight course, a tendency which increases with increased speed, and becomes strongly marked by requiring a larger helm to maintain a straight course after the limit of speed at which "squatting" commences is passed. This increased angle of rudder necessary for steering in a straight line, again increases the resistance of the vessel, which, in turn, again lessens the efficiency of the screw by increasing its losses of useful effect by slip and resistance of the screw surface to the water.

The great decrease in propelling efficiency experienced by the screw when its axis is inclined at considerable angles to the horizon is probably the principal cause why a head sea, even when the screw remains constantly immersed, so seriously lessens the speed of a screw vessel comparatively with what under the same conditions would be experienced by a paddle wheel steamer. In a head sea the pitching of the vessel keeps the axis of the screw always at a considerable angle with the horizon, and the screw as a result has its losses of useful effect by slip and by the surface

resistance of its blades so increased that its performance is lessened largely below what is due to the increased resistance of the hull.

All screw vessels should be subjected to a series of progressive trials in order to determine the speed at which "squatting" commences in each particular case, because this speed is greater and greater as the vessel is longer and longer. For economical propulsion at high speeds, the vessel's length must be increased with the speed, and the limit of speed for economical propulsion in any case can only be known by experiment. The progressive trials of the commandant's barge shows this limit of speed for a vessel of its length.

In the case of the barge, the speed at which the squatting commenced may be taken at 7.5 geographical miles per hour, the vessel's resistance in function of speed at that speed and lower speeds being in the 2.4708 power of the speed as previously determined. Assuming this law of resistance, and taking the experimental "thrusts of the screw" at the different experimental speeds, there are obtained for the speed of 7.5 geographical miles per hour, the following "thrust" deduced from trials A, B, C, D and E:

Designation of the trial.	Thrust of the screw in pounds.	Speed of the vessel in geographical miles per hour.	Thrust of the screw in pounds for the speed of 7.5 geographical miles per hour.
A	144.5433	4.2705	581.1869
B	296.4816	5.5843	614.4522
C	405.5844	6.5336	571.6124
D	472.1184	6.9142	577.0781
E	599.4837	7.5790	584.6784
			586.0000

The slip of the screw at the vessel's speed of 7.5 geographical miles per hour, taken as the mean of the slips given at the above five trials, is 27 per centum of its axial speed, or 2.774 geographical miles per hour. The corresponding number of revolutions

made by the screw per minute is 260.5315. The indicated horses-power developed by the engine and its distribution, for the speed of 7.5 geographical miles per hour, are as follows:

	Horses-power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine....	24.299831	
Horses-power expended in working the unloaded engine, or engine *per se*.....	2.292156	
Net horses-power developed by the engine and applied to the crank pin	22.007675	or 100.0000
Horses-power absorbed in overcoming the friction of the load.....	1.650576	or 7.5000
Horses-power expended in overcoming the surface resistance of the screw blades to the water.....	1.851515	" 8.4130
Horses-power expended in the slip of the screw....	4.996508	" 22.7035
Horses-power expended in the propulsion of the vessel.....	13.509076	" 61.2835
Totals.....	22.007675	or 100.0000

Taking the resistance of the water to one square foot of the immersed exterior of the hull, or wetted surface of the hull, when moving in it with the velocity of 10 feet per second, to be 0.45 pound, and at other velocities to be this quantity modified in the ratio of their squares to the square of 10, and deducing from 7.5 geographical miles per hour speed of the vesssel the mean speed of its immersed surface due to the inclination of its horizontal water lines to its longitudinal central plane, there results for that speed 12.4 feet per second, and, consequently, a surface resistance of (10^2: 0.45 :: 12.4^2:) 0.69192 pounds per square foot moving with that velocity.

As the total immersed external or wetted surface of the hull during the above performance was 533 square feet, the power expended in overcoming its resistance was ($\frac{533 \times 0.69192 \times 12.4 \times 60}{33000} =$) 8.314614 horses-power; consequently, of the 13.509076 horses-power required to propel the vessel alone at the experimental speed of 7.5 geographical miles per hour, ($\frac{8.314614 \times 100}{13.509076} =$)

61.5484 per centum were expended in overcoming the resistance of its wetted surface to the water, and the remaining 38.4516 per centum were expended in the displacement of water by the immersed solid of the hull, irrespective of the resistance of its immersed surface.

The thrust of the screw was 586 pounds, calculated as before.

Returning, now, to table No. 1, and the slips of the screw under the different experimental conditions, these slips in geographical miles per hour (line 19) will be found in experiments B, C, D and E to bear a constant relation to the thrust of the screw in pounds (line 4). If the data in *all* the experiments were exact, and if the water in all the cases came "solidly" upon the whole surface of the screw, then this constant relation would practically obtain in all the exeriments, as it theoretically should do. What is this constant relation?

The pressure which the screw exerts upon the water, is due wholly to the resistance of the vessel: if a feather were to be propelled by a large screw, the latter would exercise upon the water only the pressure due to the feather's resistance: if, retaining the same screw, a small vessel were attached to it, the pressure of the screw upon the water would increase until the resistance of the small vessel at its speed were equilibrated, and if, successive larger and larger vessels were attached and propelled at the same speed, the pressure which the same screw would exert upon the water would correspondingly become larger and larger. Indeed a screw of quite small surface, relatively, will exert sufficient pressure upon the water to equilibrate the resistance of quite a large vessel at high speed. How is this variable and increasing resistance obtained from the same screw? Simply, by the screw increasing its slip in miles per hour. The pressure will be in the ratio of the square of the slip in miles, or other absolute units of length, per hour, because a double speed of slip not only moves the water with double velocity, but moves a double mass of it in the given time. Each increase of resistance thrown upon the screw is equilibrated by an increase of its slip as above defined, and the increase of the slip of the screw will be in the ratio of the square root of the increased resistance thrown upon it.

The question may properly be asked here, how do these postulates harmonize with the so called "negative slip" of the screw? The answer is there is no such slip. All screws have, and necessarily must have, a positive slip, which may, in cases where the screw is large relatively to the resistance against it, be small, so that a slight error in the pitch or in the revolutions of the screw, or in the speed of the vessel, may make the speed of the latter *appear* to be greater than the speed of the screw, particularly if all these errors happen to be in the same direction. Negative slip supposes that the vessel moves faster than the screw propelling it, that is, the speed of the vessel per hour exceeds the product of the pitch of the screw into its number of revolutions per hour. As such a supposition is absolutely preposterous, a following current has been imagined to be generated by the vessel itself, and to move with it, so that the screw might have a positive slip in relation to this current, and a negative one with relation to the speed of the vessel. Now, if there be a following current—due to anything—the vessel moves in it as well as the screw; the same influence acts upon both, and no more affects their mutual relations than if the vessel was being propelled with the current of a tideway or a river in its favor. In order that the screw obtain any increase of propelling effect from a following current, the vessel must not be affected by it, in which case the screw would act against a more resisting medium by the amount of *vis viva* in the water of the following current due to the absolute speed of the current. In fact, however, no such following current as the one which has been invented to account for negative slip, acts in the supposed manner. The replacing current of water at the vessel's stern is vertical, not horizontal, and the water displaced by the slip of the screw also rises vertically—does not flow horizontally, as is generally supposed.

What is called the "following wave" in the case of an advancing vessel, is the capital of the ascending column of replacing water which rises above the general water level in virtue of its *vis viva*. The greater the speed of the vessel, the greater is the distance between its stern and this capital, because as the water rises with the velocity due to gravity, that velocity will be the

same in all cases of the same vessel, so that as the velocity of the vessel increases—the speed of ascent remaining constant—the space between the stern and the capital will increase as the speed of the vessel increases, instead of remaining constant if the wave were really a "following" one; in fact, the "following" wave does not follow. For propelling with maximum economy, the speed of the vessel should be such that the ascending column of replacing water will expend its entire *vis viva* on the overhanging inclined surfaces of the after body of the vessel, and thus assist the propulsion.

From the facts that the same pressure upon the water given in any case by the screw acts to propel the vessel forward and the slip or receding fulcrum of the screw aft, this pressure being known as the thrust of the screw; and that the resistance of the thrown back water constituting the slip is in the ratio of the square of its speed, there follows that the quotient of the division of the thrust of the screw in pounds by the square of the speed of the slip must be a constant theoretically, and will be a constant practically, as long as the replacing water gets "solidly" upon the whole of the screw surface, and it will reach the whole of that surface as long as the speed of the vessel does not exceed the speed which allows the ascending column of replacing water to strike the vessel's after body forward of the screw's position there. Should this not be the case, however, a portion of the screw surface would not receive "solid" water to act propulsively upon, the propelling efficiency of the screw would be decreased correspondingly to the surface thus lessened, and the slip would be correspondingly increased. In other words, there would be, in place of the supposed screw surface, less screw surface, and the quotient resulting from the division of the thrust of the screw by the speed of its slip would no longer be a constant, but would become less and less as the speed of the vessel became greater and greater, because the slip of the screw would become abnormally greater in order that the less surface might obtain the required resistance from the water.

Now experiment can determine this diminution of acting screw surface when the vessel exceeds a certain limit of speed, if a suffi-

cient number of progressive trials be made below and above that limit. The following table will illustrate, in the case of the experimental barge, the action of the foregoing principles:

Designation of the trial (Table No. 1).	Speed of the vessel in geographical miles per hour (line 18).	Speed of the slip of the screw in geographical miles per hour (line 19).	Square of the speed of the slip of the screw.	Thrust of the screw in pounds (line 44).	Quotients of the division of the thrusts of the screw by the square of the slip of the screw.
A.....	4.2705	1.3467	1.8136	144.5433	79.700
B.....	5.5843	2.0791	4.3226	296.4816	68.589
C.....	6.5336	2.4899	6.1996	405.5844	65.421
D.....	6.9142	2.7016	7.2988	472.1184	64.686
E.....	7.5790	2.9879	8.9276	599.4837	67.149
F.....	8.5247	4.0994	16.8051	924.5317	55.015
G.....	8.8525	4.9098	24.1061	1125..129	46.673

Omitting trial A, which was made at so low a speed of vessel that a slight error in the estimated pressure required per square inch of the steam pistons to work the engine *per se* or unloaded, would influence the result in a much greater degree than in the case of higher speeds; and in which the slip of the screw seems to have been too small, owing doubtless to the greater influence proportionally of even a gentle breeze, when the vessel was at a low than when it was at a moderate speed. Trials B, C, D and E, which latter was made at a slightly greater speed than 7.5 geographical miles per hour—the speed at which the "squatting" of the vessel commenced—show a very close agreement in the constants obtained from them, namely:

Designation of the trial.	Constants.
B, ...	68.589
C, ...	65.421
D, ...	64.686
E, ...	67.149
Mean, . . .	66.461

The mean 66.461 will be taken as the correct constant for all speeds of the vessel less than that (7.5 geographical miles per hour) at which "squatting" began.

For the next trial, F, the constant, owing to the considerable squatting of the vessel, fell to 55.015, and the slip of the screw rose from the mean of 27 per centum for the preceding trials to 32.4729 per centum. Now, had the water got "solidly" upon the entire screw surface in trial F, its constant would have been the same as that derived from the preceding trials, namely 66.461, applying which the speed of the slip of the screw in trial F ought to have been ($\sqrt{\frac{924.5317}{66.461}}$=) 3.7297 geographical miles per hour. The actual slip was 4.0994 geographical miles per hour, and as the pressure produced upon the water by the screw is in the ratio of the acting surface of the screw multiplied by the square of its slip in miles per hour, the acting screw surface in trial F must have been only ($\frac{3.7297^2}{4.0994^2}=\frac{13.91066}{16.80508}$=) 0.8278 of the surface in the preceding trials.

Again, for the last trial G, in which the "squatting" of the vessel was much greater than in trial F, the constant fell to 46.673 while the slip of the screw rose from the mean of 27 per centum, due to the trials made before the "squatting" of the vessel began to 35.6759 per centum. Now had the water got "solidly" upon the entire screw surface in trial G, its constant would have been the same as that derived from the trials made before the "squatting" commenced, namely 66.461, applying which the speed of the slip of the screw in trial G ought to have been ($\sqrt{\frac{1125.1129}{66.461}}$=) 4.1145 geographical miles per hour. The actual slip was 4.9098 geographical miles per hour, and, as the pressure produced upon the water by the screw is in the ratio of the acting surface of the screw multiplied by the square of its slip in miles per hour, the acting screw surface in trial G must have been only ($\frac{4.11452}{4.9098^2}=\frac{16.92911}{24.10613}$=) 0.7023 of the surface in the trials made before the commencement of the squatting.

ECONOMY TRIAL.

In addition to the progressive trials hereinbefore described, a trial of four consecutive hours was made to determine the fuel economy of the power.

This trial was conducted in precisely the same manner as the preceding one, and the same quantities were observed with the addition of the quantity of semi-bituminous coal consumed, which was very carefully weighed. The experiment commenced and ended with a full fire in the furnace. The refuse from the coal in ash and dust was not ascertained but has been taken at one-eighth of the weight of the coal consumed, as that fraction has been given by a great number of extensive experiments made with this coal in different boilers.

The vessel's draught of water and trim were exactly the same as during the progressive trials. The barge was run in opposite directions, each run of two hours' duration. There was a gentle breeze ahead during the first two hours and the same breeze astern during the last two hours. There was a very gentle swell on the water. A set of observations was taken every fifteen minutes.

The following table, No. 2, contains the data and results of the trial. In calculating the number of pounds of steam condensed in the cylinders to furnish the heat transmuted into the power developed by the expanding steam, the mechanical equivalent of one Fahrenheit thermal unit is taken at 789¼ foot-pounds:

TABLE NO. 2, CONTAINING THE DATA AND RESULTS OF THE ECONOMY TRIAL.

Date of commencing the trial	11:25 A. M., Nov. 22, 1888.
Vessel's mean draught of water in feet and inches	3 0½
TOTAL QUANTITIES.	
Length of course in geographical miles of 6,086 feet	35.6060
Time of running the course, in consecutive hours	4.
Total number of double strokes made by the pistons of the engine	84,384:
Total number of pounds of semi-bituminous coal consumed	976.
Total number of pounds of refuse from the coal	122.
Total number of pounds of combustible consumed	854.
Per centum of the coal in refuse	12.5
ENGINE.	
Steam pressure in boiler in pounds per square inch above atmosphere	148.00
Steam pressure in receiver in pounds per square inch above atmosphere	30.20

Position of the throttle valve	Wide open.
Vacuum in the condenser in inches of mercury	18.00
Height of the barometer in inches of mercury	30.52
Fraction completed of the stroke of the piston of the small cylinder when the steam was cut off	0.70
Fraction completed of the stroke of the piston of the small cylinder when the steam was released	0.95
Fraction completed of the stroke of the piston of the small cylinder when the steam was cushioned	0.875
Fraction completed of the stroke of the piston of the large cylinder when the steam was cut off	0.70
Fraction completed of the stroke of the piston of the large cylinder when the steam was released	0.95
Fraction completed of the stroke of the piston of the large cylinder when the steam was cushioned	0.875
Number of times the steam was expanded	4.337885
Number of double strokes made per minute by the pistons of the engine	351.6
Number of revolutions made per minute by the circulating pump	353.0

SPEED.

Speed of the vessel per hour, in geographical miles	8.9015
Slip of the screw per hour, in geographical miles	4.9638
Slip of the screw in per centum of its speed	35.8000

TEMPERATURE.

Temperature in degrees Fahrenheit, of the atmosphere	75.0
Temperature in degrees Fahrenheit of the injection water	47.0
Temperature in degrees Fahrenheit of the discharge water	86.0
Temperature in degrees Fahrenheit of the feed water	100.0

RATE OF COMBUSTION.

Number of pounds of semi-bituminous coal consumed per hour	244.0
Number of pounds of combustile consumed per hour	213.5
Pounds of coal consumed per hour per square foot of grate	31.6883
Pounds of combustible consumed per hour per square foot of grate	27.7273
Fraction of a pound of coal consumed per hour per square foot of heating surface	0.8328
Fraction of a pound of combustible consumed per hour per square foot of heating surface	0.7287
Number of revolutions made per minute by the blower	2294.
Air pressure in closed ash pit of boiler in column of water, fraction of an inch high	0.932

STEAM PRESSURE IN SMALL CYLINDER PER INDICATOR.

Steam pressure in small cylinder at the commencement of the stroke of its piston, in pounds per square inch above zero	141.0

Steam pressure in small cylinder at the point of cutting off the steam, in pounds per square inch above zero	125.0
Steam pressure in small cylinder at the end of the stroke of its piston, in pounds per square inch above zero	86.5
Mean back pressure against the piston of the small cylinder, including cushioning, in pounds per square inch above zero	46.0
Mean back pressure against the piston of the small cylinder, excluding cushioning, in pounds per square inch above zero	45.8
Back pressure against the piston of the small cylinder where the cushioning begins, in pounds per square inch above zero	43.5
Indicated pressure on the piston in pounds per square inch	78.00
Net pressure on the piston in pounds per square inch	74.08
Total pressure on the piston in pounds per square inch	118.075
Total pressure on the piston, of the expanding steam alone, in pounds per square inch above zero	105.00

STEAM PRESSURE IN LARGE CYLINDER PER INDICATOR.

Steam pressure in large cylinder at the commencement of the stroke of its piston, in pounds per square inch above zero	40.00
Steam pressure in large cylinder at the point of cutting off the steam, in pounds per square inch above zero	32.00
Steam pressure in large cylinder at the end of the stroke of its piston, in pounds per square inch above zero	22.00
Mean back pressure against the piston of the large cylinder, including cushioning, in pounds per square inch above zero	9.5
Mean back pressure against the piston of the large cylinder, excluding cushioning, in pounds per square inch above zero	9.2
Back pressure against the piston of the large cylinder where the cushioning begins, in pounds per square inch above zero	8.0
Indicated pressure on the piston of the large cylinder, in pounds per square inch	24.0
Net pressure on the piston of the large cylinder, in pounds per square inch	22.75
Total pressure on the piston of the large cylinder, in pounds per square inch	32.05
Total pressure on the piston of the large cylinder of the expanding steam alone, in pounds per square inch above zero	33.5

HORSES-POWER.

Indicated horses-power developed by the small cylinder	30.779295
Net horses-power developed by the small cylinder	29.232438
Total horses-power developed by the small cylinder	46.593144
Total horses-power developed by the expanding steam alone in the small cylinder	12.430100
Indicated horses-power developed by the large cylinder	29.705567
Net horses-power developed by the large cylinder	28.158402
Total horses-power developed by the large cylinder	27.022175

Total horses-power developed by the expanding steam alone in the large cylinder	28.244708
Aggregate indicated horses-power developed by the engine	60.484862
Aggregate net horses-power developed by the engine	57.390840
Aggregate total horses-power developed by the engine	73.615319
Aggregate total horses-power developed by the expanding steam alone in the engine	40.674808

ECONOMIC RESULTS.

Number of pounds of semi-bituminous coal consumed per hour per indicated horse-power	3.993726
Number of pounds of semi-bituminous coal consumed per hour per net horse-power	4.210079
Number of pounds of semi-bituminous coal consumed per hour per total horse power	3.281383
Number of pounds of what remains of the coal, after deducting its refuse, consumed per hour per indicated horse-power	3.494510
Number of pounds of what remains of the coal, after deducting its refuse, consumed per hour per net horse-power	3.683819
Number of pounds of what remains of the coal, after deducting its refuse, consumed per hour per total horse-power	2.871210

WEIGHT OF STEAM PRESENT PER HOUR IN THE CYLINDERS, CALCULATED FROM THE PRESSURES THERE AS GIVEN BY THE INDICATOR.

Number of pounds of steam present per hour in the small cylinder at the point of cutting off the steam	1,132.521387
Number of pounds of steam present per hour in the small cylinder at the end of the stroke of its piston,	1,124.753365
Number of pounds of steam liquefied per hour in the small cylinder to furnish the heat transmuted into the total horses-power developed therein by the expanding steam alone	35.168276
Sum of the two quantities immediately above	1,159.921641
Number of pounds of steam present per hour in the large cylinder at the end of the stroke of its piston	958.202275
Number of pounds of steam liquefied per hour in the small and large cylinders to furnish the heat transmuted in them into the total horses-power developed in those cylinders by the expanding steam alone	131.037614
Sum of the two quantities immediately above	1,089.239889

COST OF THE HORSE-POWER IN POUNDS OF WATER PER HOUR.

Number of pounds of steam entering the small cylinder per hour, on the supposition that the total cylinder condensation due to all causes was three times the weight of steam liquefied per hour in both cylinders, to furnish the heat transmuted in them into the total horses-power developed in both cylinders by the expanding steam alone; and that to this condensation is added the weight of steam present per hour in the large cylinder at the end of the stroke of its piston	1,351.315117

Pounds of water consumed per hour per indicated horse-power...	22.341
Pounds of water consumed per hour per net horse-power.....	23.546
Pounds of water consumed per hour per total horse-power.....	18.356

BOILER VAPORIZATION.

Pounds of water vaporized per hour in the boiler from the feed water temperature of 100 degrees Fahrenheit, being the above number of pounds of steam entering the small cylinder per hour, increased by about 2 per centum of the boiler vaporization to include the steam used in the two auxiliary engines	1,379.
Pounds of water vaporized from 100 degress Fahrenheit by one pound of coal.....	5.652
Pounds of water vaporized from 100 degrees Fahrenheit by one pound of combustible.....	6.442

EQUIVALENT PRESSURES ON THE PISTON OF THE LARGE CYLINDER ALONE.

Indicated pressure that would have been upon the piston of the large cylinder in pounds per square inch, had the indicated horses-power developed by the engine been developed in that cylinder alone.....	48.867500
Net pressure that would have been upon the piston of the large cylinder in pounds per square inch, had the net horses-power developed by the engine been developed in that cylinder alone.....	46.637510
Total pressure that would have been upon the piston of the large cylinder in pounds per square inch above zero, had the total horses-power developed by the engine been developed in that cylinder alone	59.475979
Back pressure that would have been against the piston of the large cylinder in pounds per square inch, had the indicated and total horses-power developed by the engine been developed in that cylinder alone.....	10.608479

In the foregoing table No. 2 the total horses-power developed in the small cylinder has been calculated for the total pressure on its entire piston down to the zero line: the total horses-power developed by the expanding steam alone in that cylinder has been similarly calculated for the pressure of that steam down to the zero line of pressure. But in the case of the large cylinder, the total horses-power developed therein has been calculated for the total pressure down to zero, *excluding* the cushioning, on only the annular surface remaining after subtracting the piston of the small cylinder from the piston of the large cylinder. The horses-power developed by the expanding steam alone in the large cylin-

der has been calculated for the same annular surface and for the pressure down to the zero line, *including* the cushioning, which, of course, makes this power greater than the total power, by the power due to the cushioning.

In calculating the weight of steam present per hour in the small cylinder at the point of cutting off the steam and at the end of the stroke of the piston; and in the large cylinder at the end of the stroke of its piston, the weight was first calculated for the cylinder capacity, plus the space in the clearance and steam passage at one end of the cylinder; and from this weight there was deducted the weight of steam in the cylinder and in the clearance and steam passage at the point where the cushioning began, using the back pressure at that point for determining this weight.

To the weight of steam calculated for the small cylinder at the point of cutting off the steam, no addition is to be made: it is the entire weight accounted for by the indicator at that point; but to the weight of steam calculated for the small cylinder at the end of the stroke of its piston, there must be added the weight of steam liquefied in that cylinder to furnish the heat transmuted into the horses-power developed by the expanding steam alone; the sum of the two quantities constitutes the weight of steam accounted for by the indicator at the end of the stroke of the piston of the small cylinder. In the case of the large cylinder, there must be added to the weight of steam present at the end of the stroke of the piston, the weights of steam liquefied in both the small cylinder and the large cylinder to furnish the heat transmuted into the power developed by the expanding steam. The sum of the three quantities constitutes the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder.

The number of pounds of steam liquefied per hour in the cylinder to furnish the heat transmuted into the total horses-power developed by the expanding steam alone, is obtained by multiplying the mechanical equivalent of the Fahrenheit unit of heat (789¼ foot-pounds) by the number of minutes (60) in an hour, and that product by the total horses-power developed by the expanding steam alone, and dividing the latter product by the

latent heat of the mean pressure of the expanding steam down to zero.

The steam accounted for by the indicator does not include the steam expended in the cylinders of the circulating pump and of the blower. This quantity can only be estimated, and certainly will not exceed two per centum of the quantity vaporized in the boiler.

The weight of coal consumed includes, of course, the weight consumed for producing the steam expended in the cylinders of the circulating pump engine and of the blowing engine. The power developed by the cylinder of the circulating pump engine should be added to the power developed by the cylinders of the main engine, the sum being the power exerted by the engine, and this aggregate power should be divided into the weight of coal consumed per hour in order to obtain the true cost of the horse-power in pounds of coal consumed per hour. The cost of the horse-power in pounds of coal and of combustible consumed per hour, as given in table No. 2, has been obtained by deducting one per centum from the weight of coal consumed per hour, as the quantity used in producing the steam expended in the engine of the circulating pump, and dividing the remainder by the indicated, net, and total horses-power developed by the main engine.

The power developed by the blowing engine should not be included as a part of the engine power. It is wholly consumed in producing an artificial draught for the boiler, and is no more chargeable to the main engine than the heat passing in the gases of combustion from the chimney into the atmosphere when natural draught is used. It is merely the cost of the additional draught over what the natural draught would cost in heat, and is due to the system.

The weight of water vaporized per hour in the boiler has been calculated in the table by adding to the weight of steam accounted for by the indicator at the end of the stroke of the piston of the large cylinder, calculated from the pressure there, three times the weight of steam liquefied in the cylinders to furnish the heat transmuted into the power developed by the expanding steam alone, as

experiment has shown in many cases, that this is about the proportion of cylinder condensation due to all causes; additionally, 2 per centum of the weight of water vaporized in the boiler is allowed for the steam used in the two auxiliary engines. The division of this final weight of water by the coal gives the economic vaporization by the latter.

There remains only to add the "distribution of the power" during the performance in table No. 2, as follows:

	Horses-Power.	Per centum of the net horses-power.
Indicated horses-power developed by the engine.....	60.484862	
Horses-power expended in working the unloaded engine, or engine *per se*.....	3.094022	
Net horses-power developed by the engine and applied to the crank-pin.....	57.390840	or 100.0000
Horses-power absorbed in overcoming the friction of the load.....	4.304313	" 7.5000
Horses-power expended in overcoming the surface resistance of the screw blades to the water.....	4.550877	" 7.9296
Horses-power expended in the slip of the screw.....	17.375763	" 30.2762
Horses-power expended in the propulsion of the vessel.	31.159887	" 54.2942
Totals.....	57.390840	or 100.0000

The thrust of the screw, calculated as hereinbefore described, was 1,138.8484 pounds.

The horses-power expended in overcoming the immersed external surface resistance of the hull to the water, calculated as hereinbefore described, was 13.901037 horses-power, consequently, of the 31.159887 horses-power required to propel the vessel alone at the experimental speed of 8.9015 geographical miles per hour, $(\frac{13.901037 \times 100}{31.159887} =)$ 44.6120 per centum were expended in overcoming the resistance of its wetted surface, and the remaining 55.3880 per centum were expended in the displacement of water by the immersed solid of the hull, irrespective of the resistance of its immersed surface.

As the wetted surface is a constant quantity at all speeds of

vessel, and as its resistance varies as the squares of the speeds of the vessel, the expenditure of power to overcome this surface resistance at different speeds will be in the ratio of the cubes of the speeds, and, consequently, is unaffected by the "squatting" of the vessel. Hence, the increased resistance of the hull above what is due to the square of the speed, which obtains after squatting commences, appertains wholly to the immersed solid of the hull, irrespective of its surface. In the experimental speed of 7.5 geographical miles per hour, above which "squatting" commenced, the power required to overcome the resistance of the immersed solid of the hull irrespective of its surface, was 5.194462 horses, and when the speed was increased above that limit to 8.9015 geographical miles per hour, the vessel gradually squatting more and more as the speed became greater and greater than 7.5 miles per hour, the resistance of the immersed solid of the hull irrespective of its surface became 17.258850 horses-power, consequently, the horses-power increased in the 7.0089 power of the speed, instead of the theoretical 3d power of the speed. The resistance of the immersed solid of the hull irrespective of the resistance of its wetted surface, as measured by the portion of the thrust of the screw applied to overcoming it, was, of course, in the 6.0089 power of the speed.

From the foregoing will be perceived that a vessel of given linear dimensions should be modeled wholly with reference to the speed at which it is to be used. For all the speeds below the speed at which "squatting" commences, the model should be fuller than for speeds above that limit; and the vessel's coefficient of displacement should become less and less the more that limit is exceeded.

Previously to the application of steam power to the propulsion of vessels, nothing could be known of the dynamical problems of naval architecture because the propelling power could be neither geometrically applied nor measured. The vessel moving under the oblique pressure of its sails, was always more or less careened, and its trim was modified by the excessive leverage on the hull of the center of pressure of the sails. The statical problems had all been perfectly solved by eminent mathematicians,

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TABLE NO. 1, CONTAINING THE UNCORRECTED DATA AND RESULTS OF THE PROGRESSIVE T
APPROPRIATED TO THE USE OF THE COM

These trials were made during November, 1888, in the Hudson river near New York City, the Barge when at rest bein

Number of line.		A.	
1	Date of commencing trial	2:45 P. M., Nov. 16	1:
	TOTAL QUANTITIES.		
2	Length of course in geographical miles at 6,086 feet	3.	
3	Time of running the course, in minutes and seconds	42:9	
4	Total number of double strokes made by the pistons of the engine	6004.	
	PRESSURES.		
5	Steam pressure in boiler, in pounds per square inch above atmosphere	152.7	
6	Steam pressure in receiver, in pounds per square inch above atmosphere.	4.0	
7	Height of the barometer, in inches of mercury	30.17	
8	Vacuum in the condenser, in inches of mercury	9.83	
9	Fraction used of the throttle valve opening	0.0625	
	TEMPERATURES.		
10	Temperature in degrees Fahrenheit, of the atmosphere	71.7	
11	Temperature in degrees Fahrenheit, of the injection water	52.0	
12	Temperature in degrees Fahrenheit, of the discharge water.	62.7	
13	Temperature in degrees Fahrenheit, of the feed water	77.3	
	AUXILIARY ENGINES.		
14	Number of revolutions made per minute by the circulating pump	224.	
15	Number of revolutions made per minute by the blower	Not used.	
16	Height, in fractions of an inch, of the column of water equilibrating the air pressure in the ash pit of the boiler	.....	
	SPEED.		
17	Number of double strokes made per minute by the pistons of the engine	142.4437	
18	Speed of the vessel per hour in geographical miles of 6,086 feet	4.2705	
19	Slip of the screw in geographical miles per hour	1 3467	
20	Slip of the screw in per centum of its speed	23.9750	
	STEAM PRESSURE IN CYLINDERS.		
21	Indicated pressure on the piston of the small cylinder in pounds per square inch	17.123	
22	Net pressure on the piston of the small cylinder in pounds per square inch	13.203	
23	Indicated pressure on the piston of the large cylinder in pounds per square inch	3.073	
24	Net pressure on the piston of the large cylinder in pounds per square inch	1.823	
25	Equivalent indicated pressure on the piston of the large cylinder alone in pounds per square inch	8.532054	
26	Equivalent net pressure on the piston of the large cylinder alone in pounds per square inch	6.032302	
	HORSES-POWER.		
27	Indicated horses-power developed by the small cylinder	2.737400	
28	Net horses-power developed by the small cylinder	2.110722	
29	Indicated horses-power developed by the large cylinder	1.540932	
30	Net horses-power developed by the large cylinder	0.914129	
31	Aggregate indicated horses-power developed by the engine	4.278 132	
32	Aggregate net horses-power developed by the engine	3.024851	
	DISTRIBUTION OF THE INDICATED HORSES-POWER.		
33	Indicated horses-power developed by the engine	4.278332	
34	Horses-power expended in working the unloaded engine, or engine *per se*	1.253481	
35	Net horses-power developed by the engine and applied to the crank pin	3.024851	
36	Horses power absorbed in overcoming the friction of the load	0.226864	
37	Horses-power expended in overcoming the surface resistance of the screw blades to the water	0.302606	
38	Horses-power expended in the slip of the screw	0.508268	
39	Horses-power expended in the propulsion of the vessel	1.897113	
	DISTRIBUTION OF THE NET HORSES-POWER.		
40	Per centum of the net horses-power absorbed in overcoming the friction of the load	7.5000	
41	Per centum of the net horses-power expended in overcoming the surface resistance of the screw blades to the water	10.0040	
42	Per centum of the net horses-power expended in the slip of the screw	19 7784	
43	Per centum of the net horses-power expended in the propulsion of the vessel	62.7176	
44	Thrust of the screw, in pounds	144.5433	

E TRIALS MADE BY A BOARD OF UNITED STATES NAVAL ENGINEERS WITH THE STEAM BARGE OMMANDANT OF THE NEW YORK NAVY YARD.

being maintained at 3 feet ½ inch mean draught of water, with the rabbet of the keel parallel to the surface of the water.

	B.	C.	D.	E.	F.		G.
	1:54 P. M., Nov. 16.	12:38 P. M., Nov. 21.	1:18 P. M., Nov. 21.	1:46 P. M., Nov. 21.	2:18 P. M., Nov. 21.	2:40 P. M., Nov. 21.	11:30 A. M., Nov. 22.
	3. 32:14 6264.	3. 27:33 6304.	3. 26:2 6348.	3. 23:45 6364.	3. 21:7 6760.	3. 21:2 6936.	3. 20:20 7096.
	153.3 6.0 30.17 13.00 0.1250	151 7 11.0 30.64 10.83 0.1875	150.7 13.0 30 64 11.17 0.2500	150.0 15.0 30.64 12.33 0.3125	150.0 25 0 30.64 14.67 0.3750	151.7 28.3 30.57 16.33 0.5000	138 3 31.3 30.51 17.50 1.0000
	62.0 51.7 71.3 79.3	68.0 46 7 70.7 73.0	73.0 46.0 72.7 70.0	72.3 46.0 76.0 70.0	70.3 46.0 82.7 72.3	69.0 46.0 88.0 80.7	65.0 47.0 88.0 88.0
	219. Not used.	220. 564. 0.125	220. 606. 0.125	257. 568. 0.125	247. 1259. 0.292	252. 1529. 0.438	300. 2130. 0.750
	194.3330 5.5843 2.0791 27.1306	228.8203 6.5336 2.4899 27.5944	243.8412 6.9142 2.7016 28.0956	267.9579 7.5790 2.9879 28.2758	320 1263 8.5247 4.0994 32.4729	329.7623 8.5578 4 4463 34.1915	3?8.0901 8 8525 4.9098 35.6759
	26.483 22.563 6.307 5.057 14.750154 12.250402	33.650 29.730 8.553 7.303 19.281095 16.781535	38.420 34.500 9.737 8.487 21.985520 19.486087	47.353 43.433 12.027 10.777 27.123804 24.624053	65.640 61.720 19.283 18.033 40.2?9960 37.710207	....	74.747 70 827 24 467 23.217 48 297?98 45.797647
	5.776023 4.921159 4.314665 3.459531 10.090688 8.380690	8.641613 7.634923 6.889548 5.882658 15.531160 13.517581	10.514283 9.441508 8.368157 7.285159 18.872440 16.726667	14.240632 13.061757 11.344925 10.165815 25.585557 23.227572	23.583130 22.174941 21.736?9? 20.322029 45.314029 42.496970	....	29 276689 27 741?15 30 058?9? 28.52 114 59 135484 56.264429
	10.090688 1.709998 8.380690 0.628552 0.768398 1.894731 5.089009	15.531160 2.013579 13.517581 1.013819 1.254387 3.104197 8.145178	18.872440 2.145773 16.726667 1.254500 1.517993 3.920509 10.033665	25 585557 2.357985 23.227572 1.742068 2.014410 5.505608 13.965486	45.314029 2.817059 42.496970 3.187273 3.434891 11.649590 24.225216	....	59.335484 3.071055 56.264429 4.219812 4.450282 16 979700 30.614615
	7.5000 9.1687 22.6083 60.7230	7.5000 9.2797 22.9641 60.2562	7.5000 9.0753 23.4387 59.9860	7.5000 8.6725 23.7029 60.1246	7.5000 8.0827 27.4128 57.0045	....	7.5000 7.9096 30.1784 54.4120
	296.4816	405.5844	472.1184	599.4837	924.5317	.....	1125.1129

and there remained nothing to add; to that extent, therefore, naval architecture was as exhausted a science as geometry. But the, if possible, more important dynamical problems still await the necessary experimental data which alone can solve them. Abstract science is of but little use in their solution, hence all steam vessels should be subjected to a complete set of accurately conducted progressive trials, and all the data measurable should be measured with proper instruments by experts, the results in each case being carefully analyzed and reported with all the drawings and dimensions necessary to a complete understanding of the case. This would be interrogating nature on the scale of practice, and the money value of her answers would be so great that the cost of the questioning would sink, comparatively, into insignificance. Without such experimentation and reporting, the designing of vessels with the view of obtaining their maximum economic performance relatively to power applied, is but little better than guess-work.

PASSED ASSISTANT ENGINEER ASA M. MATTICE, U. S. N.—The author of the paper, attributes "squatting", or change of trim of a vessel in motion, solely to the cavity caused by the forward motion of the vessel not being filled with sufficent rapidity. This is one cause, but not the only one. A vessel might be made with an after body of sufficient length and of proper form to allow the cavity to be filled with sufficient rapidity, but "squatting" might still result from the hollow of the bow or displacement wave occurring somewhere along the after part of the vessel. When the hollow of the bow wave coincides with the hollow caused by a full after body the "squatting" will be a maximum. In the commandant's barge I think that a change in the form of the fore body by which the amplitude of the bow wave would be decreased would give a greater speed and less "squatting" with the same engine power, as the hollow of the wave would be brought further forward. There has been considerable change of late years in the form of fore bodies of small craft intended for speed. A few years ago the "wave line" was the only recognized proper form. It took the water from a state of rest and moved it aside

with a uniform acceleration and then brought it to rest with a uniform retardation. Experience found, however, that a water line having a less angle of greatest obliquity would create a bow wave of less amplitude, as the maximum speed of displacement is less. Now a water line having outward curvature only can have a smaller maximum angle of obliquity than a line of contrary flexure, for the same beam and length of fore body. An examination of the lines of some of the more successful of recent high-speed small craft will show that the builders appreciate this fact. In some cases there is only a slight suspicion of concavity of the lines, while in others the lines start out straight, or slightly convex, from the stem. The lines of the Herreshoff yacht "Leila", shown in Chief Engineer Isherwood's report on that vessel in 1881, are a good example. Not only should the coefficient of displacement be less for the higher speeds, as indicated by the author, but due attention must be paid to the forms of the lines of both the fore and after bodies; the former to decrease the wave-making tendency, and the latter to allow the water to be replaced with sufficient rapidity. The coefficient of displacement can be made very small by making very hollow entrance lines, but such a hull would have a large wave-making resistance, and would "squat" more than a fuller vessel with better lines.

I cannot agree with the author about the "following wave". We all know that if a square-stern boat is dragged through the water eddies can be plainly seen where the surface water turns sharply around the corner of the stern and follows the boat until lost in the dead water at the middle of the wake. The phenomenon is also sometimes seen of a boat being towed behind a bluff-stern vessel with a slack tow rope; that is to say, the boat is carried along by the eddying water following the vessel, with little or no pull on the tow rope. Even with the best formed vessels a film of water of greater or less thickness, according to circumstances, is set in motion by the hull, and the "following wave", being made up in part of this water, must "follow" to some extent. This is greatest in shallow water, where the small depth of water under the vessel causes a greater forward current to be induced. This was exemplified in some of the preliminary trials

of the "Vesuvius". In running at full speed in six fathoms of water there was a large following wave which appeared constantly to be about to come on board, while in three fathoms of water the edge of the wave did actually come over the stern.

The theory of so-called "negative slip" is not at all incompatible, in my mind, with the theory of positive slip. In order to produce a given thrust an amount of water must be moved aft from its primary position with a given velocity. This water need not be at absolute rest when taken by the propelling instrument. It is the *change* in velocity which produces the thrust; not the actual velocity. If the proper amount of water is moving forward with a given velocity the propelling instrument need give it an absolute motion in a backward direction only sufficiently great to produce the necessary reaction; and thus, while actually producing a certain *relative* sternward velocity of the water apparently produces a less, because the actual backward movement of the water *is* less. It is not easy to find a strictly analogous case to illustrate the theory of negative slip, but for want of a better the following may perhaps answer the purpose. In some forms of compressed-air refrigerating machinery the air is first compressed by a steam engine, and then, after the heat of compression has been abstracted, it is allowed to expand to a certain extent in a cylinder, which forms a part of the engine, thus giving back part of the work which was originally done upon it. Now, if we measure the work done in the steam cylinder and in the compression cylinder we will find, making due allowance for loss by friction, that the work done in the compresser is greater than the work done by the steam engine, or the engine has "negative slip", as we might say. The solution of this apparent anomaly is to be found in the air-expansion cylinder when the the compressed air is made to help "work its passage", so to speak. In the same way with a vessel having a badly formed run, some of the water which has been uselessly set in motion comes around behind the propeller and "helps push". Of course, in the case of the vessel, there is no gain by the "negative" slip, but quite the contrary, because we have set a large quantity of water in motion by the bad shape of the stern

and have only utilized the *vis viva* of a portion of this water to assist in propulsion.

Another point on which I cannot agree with the author is the theory that the water displaced by the slip of the screw rises vertically instead of flowing horizontally. The propelling force is the horizontal component of the reaction caused by the displacement of the water, and if the water rises vertically there cannot be any horizontal component, therefore no propulsion. Towards the surface is undoubtedly the direction of least resistance, but the water is set in motion, or rather is acted on by a force, in practically a horizontal direction, and it must move in that direction unless acted upon by external forces. An external force exists in the difference of pressure of the surrounding water in various directions. The resultant of these pressures tend to force the moving particle of water upwards, and it will, therefore, take a direction resultant upon the forces acting on it, or an obliquely upward direction. If we project a ball into the air it has, after leaving the hand, certain external forces acting upon it—the resistance of the air and the attraction of gravitation, the former of which may be neglected. The direction of least resistance of the latter is downward, but we do not deduce from this the theory that the ball will fall immediately to the ground, but we know that it will take a certain resultant direction, finally reaching the ground, but at a distance from the hand. In the case of the moving particle of water the tendency to rise exists only as long as the motion imparted by the propelling instrument exists, so that when this motion is destroyed by fluid friction the vertical motion ceases, and it is doubtful if more than a very small portion of the water ever reaches the surface.

I think that the necessity of making careful progressive trials of fast screw vessels cannot be too strongly urged, and besides the usual instruments of observation there should be means of measuring the actual thrust so as to eliminate the various losses and tell exactly what force is required to overcome the resistance of the vessel. In the original design of the "Yorktown's" machinery the Bureau of Steam Engineering contemplated the introduction of a permanent thrust-dynamometer. The thrust

bearing was of the "horse-shoe" pattern with a very small fore-and-aft play in the rods on which the horse-shoes were strung. These rods were connected through a system of levers with a hydraulic diaphragm of the type used by Mr. Emery in the Watertown testing machine and other delicate weighing apparatus. From this instrument a very small pipe led to a delicate gauge in the engine room. Another small diaphragm near the gauge was fitted to be put under pressure by a screw and hand-wheel. Ordinarily there would be no pressure in this hydraulic system, but when a reading of the thrust was desired the hand-wheel would be turned and pressure exerted on the small diaphragm, thus putting the system under pressure and moving the thrust-block aft from its stops an infinitesimal distance. The reading of the gauge would then show the thrust *plus* the friction of the propeller-shaft and thrust-block. Upon slowly slacking back the hand-wheel the gauge reading would fall to some extent, and would show the thrust *minus* the friction. The mean of the two readings would thus give the actual thrust as near as the mechanism could be made to indicate. This instrument was, unfortunately, never constructed, as the machinery was built on plans furnished by the contractor, and a valuable opportunity to increase our knowledge of propulsion thereby lost.

ASSISTANT ENGINEER H. W. SPANGLER, U. S. N.—It may be of interest in connection with this paper to note some experiments made at Stevens Institute to determine the friction of the load. In all of Mr. Isherwood's work the power used in overcoming the increased friction due to the load has been taken as 7½ per cent. of the power remaining after subtracting from the indicated horse-power that necessary to run the engine at the speed, unloaded. The experiments referred to were made on engines running at constant speed, and it was found that the difference between the power indicated and that absorbed by a friction brake or sent through a transmitting dynamometer was practically *constant.* Mr. Barrus, at the XIV meeting of the A. S. M. E., cited an engine running at constant speed in which the engine unloaded required 6.1 horse-power to run it; at 44.5 I. H. P. the brake power was 6.1 less; at 34.6 I. H. P. the brake power was

5.6 less; and at 25.3 I. H. P. the difference was 6.3. Numerous other examples could be referred to in which, at constant speed, the friction of the engine unloaded was practically the same as when loaded. It is true that both these examples are from non-condensing engines, and the power was taken off by belt or brake, but as the engine dealt with by Mr. Isherwood has a separate circulating pump, it seems probable that some such condition of affairs holds in this engine if we omit the friction on the thrust-block. At 7½ knots the thrust is 586 lbs., according to Mr. Isherwood. This is distributed over a ring 5¾ inches outside diameter and 1¾ inches inside diameter. The friction due to the load is not likely to be much greater than that due to this pressure, and the power absorbed in overcoming it falls far short of the 7½ per cent. used by Mr. Isherwood.

The question of negative slip is a peculiar one. It does not seem possible that any ship could move faster than the screw driving it, but a statement that this can not be in the face of a series of trials like those of the "Iris", in which six times the slip is given as negative and ten times as positive, requires some explanation. Mr. Isherwood puts the trouble down to errors of observation. There are only three quantities to be used in determining the amount of slip—the time of moving over a given distance, the number of revolutions in that same time and the pitch of the screw. Taking any set of careful observations such as were made during the trials of the "Iris," it is almost incredible that any errors or combination of errors could make a difference of 5.33 per cent. in the slip, as it must have been positive, and the results are certainly negative as calculated from the observations. It cannot be doubted that the observations are reasonably correct. Our slip is not observed, however, it is calculated, and may not our method of calculation be wrong? Many of the screws in which negative slip has shown itself have been either of expanding pitch or adjustable blades. A propeller 20 feet in diameter, we will suppose, turns once per second; its axis is horizontal, and its pitch expands from 22 to 25 feet. A block allowed to move freely horizontally without friction rests against the leading edge of a blade, and by the time the

blade of the propeller has passed the block it will be moving at 25 feet per second, and not at 22½, the mean pitch. It seems clear that the slip of a propeller of this kind should be calculated from the greatest pitch and not the mean pitch. This will in most cases make the slip positive, as it should be, or at least brings it within the limits that can safely be due to errors of observation.

The explanation of what is called the "following wave", seems to me to be unsatisfactory. The method of reasoning used can be put in a few words: 1st. The ship moving along tends to vacate certain space. 2d. This space is filled with water from below *under the influence of gravity*. 3d. This water can rise "above the general water level in virtue of its *vis viva*". The first is without doubt true; the second, modified somewhat, is true also; the third, taken in connection with the second, can not be true. If it were true the problem of perpetual motion would be once more solved, for we would get work for nothing. A hollow, empty, open-top vessel is fastened in a second vessel of water so that the open-top is just above the level of the water. Here is our first condition exaggerated. A hole is made in the bottom; here is our second condition, as the water tends to fill the vessel by gravity. If our third condition is true, our water inside must rise above that outside by its *vis viva*, and this is simply not so. There is one other matter that I would like to refer to, if I am not taking up too much space, and that is the use of the constant 789¼ for the mechanical equivalent of heat. This is based on experiments made in France, and reported on by a committee of of the Institute, February 13, 1865. It is to be remembered that M. Tresca was one of the experimenters whose work was reported on by this committee. The number in French units was 433, the equivalent of 789¼. To show the amount of confidence M. Tresca put in this value, I quote from his work in *Comptes Rendus*, vol. xcvii, p. 223, June 8, 1874—"l'effet calorifique produit aura pour expression $A\,b \times d \times c\,(t - t_0)$ et, en multipliant cette quantité de chaleur développée par 425, on aurait pour valeur de son equivalent en travail, $T' = 425\ A\,b\,d\,c\,(t - t_0)$", showing that M. Tresca did not put sufficient faith in the results

of his own experiments to use this constant (the equivalent of 789¼).

ASSISTANT ENGINEER T. W. KINKAID, U. S. N.—Mr. President: As reference has been made to the vertical direction of the water of replacement in the case of the action of the screw, I would like to recall an incident which forcibly illustrated the point, and which was described in a paper on "Twin Ship Propulsion", by Mr. G. C. Mackrow, Trans. Inst. Naval Architects, 1879. A twin ship consisting of two hulls joined together, somewhat after the fashion of a catamaran, and having a paddle wheel located between the hulls, was under trial at the dock. When the paddles began to act, it was expected, very naturally, that a very strong current would be induced between the hulls fore and aft. The expectation was only partially realized; for, while the water ran like a mill-race between the runs of the two hulls, yet, to the surprise of everybody, the surface of the water forward of the paddles was quite tranquil, floating chips and the like not being disturbed in the least. Investigation showed that the supply of water to the paddles came from under the hulls, its direction having, of course, a decided vertical component.

PASSED ASSISTANT ENGINEER G. W. BAIRD, U. S. N.—I did not understand that part of Mr. Isherwood's paper referring to the following current in the same way as does Mr. Mattice. By a following current I understand a body of water having a motion of translation; that is, its particles moving in the same direction with, sensibly, the same velocity. There is, without doubt, an eddy along the sides and at the ends of a body moving in the water, and that eddy is augmented as the velocity of the floating body is increased, and also as its bluntness is exaggerated.

I had the honor, some 18 years ago, to assist Mr. Isherwood in his experiments at Mare Island, on the screw propeller. This same question of relative increase of speed and power arose, for it was found that the established ratio increased when a speed of nine knots was exceeded. Mr. Isherwood placed a straightedge on the cabin floor and brought it to a horizontal by a spirit level which he placed on top of it; the vessel remained level until a velocity of nine knots was exceeded, when she began to rise at the bow. The

lines of the vessel were such as to cause that rise, a species of model which had been handed down to us from the sail period; a form adapted to resisting the pressure of the sails, the centre of whose pressure was above the tops, ordinarily. Any form of vessel which will alter its draught when moved through the water must, necessarily, introduce additional resistance. The first model I have any recollection of which seemed to take cognizance of these laws, was made by the late Chief Engineer Stimers, U. S. Navy, some time previous to 1864. It had been sent to the Department by its inventor, and I saw it first, in that year, among a pile of rejected models, in the cock-loft of the Winder building. It embraced the essential features of the great U-bow section, so admirably demonstrated at a later period by the distinguished marine architect, Mr. E. J. Reed. Verily, "the stone which the builders rejected, the same has become the head of the corner".

A vote of thanks was accorded Mr. Isherwood for his Paper.

II.

DUTY AND CAPACITY TESTS OF THE HIGH SERVICE PUMPING ENGINES, WASHINGTON, D. C.

By Passed Assistant Engineer G. W. Baird, U. S. Navy.

The tests herein described were made in August, 1888, for the Commissioners of the District of Columbia, on the High Service pumping engines at the U street station, in accordance with the contract entered into between the Commissioners and the Holly Manufacturing Company, of Lockport, New York.

The following description of the plant and of the tests, together with the results, are embodied in my report to the Commissioners:

THE PLANT.

The plant is situated on a reservation between U and S and

Sixteenth and Seventeenth streets, in this city, its purpose being to receive water from the 20-inch low service main, and to deliver either into the high service main or into the stand pipe at the head of Sixteenth street. The high service main referred to supplies all that portion of the city shown in Plate I; while the stand pipe, which holds 270,919 gallons of water, supplies that portion of the city beyond the boundary, and what is known as Columbia Heights.

The plant consists of two engines, according to the patents of H. F. Gaskill, and two horizontal, externally fired boilers, which were built and erected by the Holly Manufacturing Company. Either boiler may be used to drive either engine. The plants are duplicate.

THE BOILERS.

The boilers are designated as the North and the South boiler, and the engines as the East and the West engine. The boilers are much larger than is essential to drive the engines; in fact one boiler is ample to run both the engines at the contract speed. The boilers are set in brick work, with the usual cast iron fronts, having closely fitting furnace doors and ash pit doors. The furnace is at the front end. The gases of combustion pass first under the bottom of the boiler, then returning through tubes, and return again over the top of the shell to the back end, to the smoke flue. Almost the whole of the external surface of the boiler is thus enveloped in the heated gases of combustion. The superheating surface on the top of the shell is inefficient, for the reason that it is quickly covered with soot.

The principal dimensions of the boilers are as follows:

Number of boilers	2.
Diameter of the shells	60 inches.
Length of the shells	14 feet.
Number of tubes in each boiler	54.
External diameter of the tubes	3¾ inches.
Thickness of the tubes	⅛ inch.
Length of the tubes	14 feet.
Least distance between the tubes and the shell, at the sides	2½ inches.
Least distance between the tubes and the bottom of the boiler	5½ inches.
Spacing of the tubes, vertically	5 inches.

Spacing of the tubes, laterally	5¼ inches.
Diameter of steam drums	2 feet 9 inches.
Height of the steam drums	3 feet 6 inches.
Number of furnaces to each boiler	1.
Width of the furnace grates	5 feet 6 inches.
Length of the fuanace grates	5 feet 6 inches.
Area of grate surface in each boiler	30.25 square feet.
Area of grate surface in both boilers	62.5 square feet.
Aggregate cross area, for draught through the tubes, in each boiler	3.71 square feet.
Aggregate cross area, for draught over the bridge wall of each boiler	4.03 square feet.
Area for draught through the chimney, at smallest contraction	3.82 square feet.
Water heating surface in the shell of each boiler	63 square feet.
Water heating surface in the tubes of each boiler	742.16 square feet.
Water heating surface on the heads of each boiler	14.93 square feet.
Aggregate water heating surface in each boiler	820.09 square feet.
Water room in each boiler	139.28 cubic feet.
Superheating surface in the shell of each boiler	54.978 square feet.
Superheating surface in the drum of each boiler	15.119 square feet.
Aggregate superheating surface in each boiler	70.097 square feet.
Steam room in each boiler	69.6 cubic feet.
Ratio of water heating to grate surface	27.11 to 1.
Ratio of grate surface to area over the bridge wall for draught	7.5 to 1.
Ratio of grate surface to area through the tubes for draught	8.15 to 1.
Ratio of grate surface to area through the chimney for draught	7.92 to 1.
Height of chimney above the grate	88 feet 4 inches.
Extreme length of the brick work of each boiler	20 feet 6 inches.
Extreme width of the brick work of each boiler	8 feet 7 inches.
Extreme height, above the floor, of the brick work of one boiler.	11 feet 7 inches.
Width of top edge of grate bars	1⅝ inches.
Width of air space between grate bars	⅝ inch.

COAL.

The fuel consumed during the test was a good quality of George's Creek Cumberland (semi-bituminous) coal, of the kind regularly shipped by the contractor, and was burned just as received.

A special analysis by A. S. McCreath gave the following:

Hygrometric water,	1.01
Volatile matter, other than water,	13.92
Fixed carbon,	82.96
Ash,	3.11

THE ENGINES.

There are two separate engines shown on Plate, each having four steam cylinders, and two water cylinders. The engines are horizontal, compound, rotative engines, and are in pairs; each pair acting on a crank placed at right angles on a single shaft. There is a fly-wheel on the middle of the shaft.

The high pressure cylinders are placed on top of the low pressure cylinders, with very short steam passages between them.

The piston rod of the high pressure cylinder is connected, through links, to the upper end of a working beam, and the two piston rods of the low pressure cylinders are connected similarly to the lower end of the same beam. The movements of the pistons of each pair are, therefore, coexisting and alternate. From the cross-heads of the high pressure cylinders the connecting rods extend to and work the cranks, and from the cross-heads of the low pressure cylinders the pumps are worked.

The rock shafts (the beam center), extending from the beams, carry arms which work the air and feed pumps.

The valves are actuated by eccentrics on revolving shafts, which receive their motion from the main shaft through the intervention of miter gearing. The opening of the high pressure steam valves (which are poppets) is effected by the intervention of reversed lifter toes with an intervening lever. By a unique arrangement the one toe slides off the other, effecting a trip cut-off. A dash pot is employed to cushion the valve in its fall.

The intermediate valves and the exhaust valves of the low pressure are gridiron slides. The valves are readily accessible for repairs. The cut-off (the h. p. steam) valves are momentarily adjustable.

The steam jackets, which cover the heads and sides of all the steam cylinders, drain into a kind of feed water heater, the supply of steam being regulated by valves, without the interposition of steam traps.

The cross head guides are single rectangular bars and the cross heads are of the locomotive pattern.

The links and connecting rods are fitted with the ordinary gibs and keys, and the pillow blocks with the ordinary caps and bolts.

The beam centers are carried on cast iron pedestals bolted to the sole plates, and are braced, from the steam cylinder on the one side and the water cylinder on the other, by wrought iron braces having collars and keys. The engines are mounted on brick foundations.

The water end is of the usual form, except that it has a plunger (after the Worthington style) which is packed by means of a stuffing box and gland; this packing is reached only by removing the cylinder head of the pump. The water valves are of bronze, with rubber seats. The net opening of each valve is 1.35 square inches and its weight is 15 ounces.

There is an air chamber to each water end. A No. 8 Westinghouse air pump is provided to supply air to these chambers when necessary.

There is a surface condenser for each engine. The condensers are cylindrical in form, with their axes horizontal, they each contain 128 brass tubes 1 inch outside diameter, and 7 feet 5 inches in length. The circulating water passes through the tubes and the circulation is derived by a shunt from the main suction.

There are four single acting air pumps on each engine, having a diameter of 10 inches and a stroke of 12 inches. The aggregate piston displacement of the air pumps, per revolution, is 2.18 cubic feet. The air pump valves are 8 inches diameter. There are four single acting piston feed pumps on each engine, worked from arms on the beam center; they receive their water from the feed water heater above described, which receives its heat from the drains from the jackets.

The main shaft is carried on pillow blocks, which are cast on the discharge chambers of the main pumps.

The fly wheel, which is on the middle of the main shaft, runs between the two pump cylinders.

The pumps receive their water from a 20-inch main, and discharge into the high service mains against a head of about 57 feet ordinarily.

The following are the principal dimensions of one engine.

Number of high pressure cylinders	2.
Diameter of the high pressure cylinders	12 inches.
Diameter of the high pressure piston rods	1⅞ inches.

Net area of each high pressure piston	111.71
Stroke of the high pressure pistons	22 inches.
Displacement of the high pressure piston per stroke,	1.4222 cubic feet.
Space in clearance and steam passages at one end of high pressure cylinder	.035 cubic feet.
Ratio of clearance to piston displacement	2.461 per cent.
Area of steam port of high pressure cylinder	7⅛ square inches.
Area of exhaust port of high pressure cylinder	12¼ square inches.
Clearance of high pressure piston	⅛ inch.
Number of low pressure cylinders	2.
Diameter of the low pressure cylinders	24 inches.
Diameters of the (two) low pressure piston rods	2¼ inches.
Net area of each low pressure piston	448.41 square inches.
Stroke of the low pressure pistons	22 inches.
Displacement of each low pressure piston for one stroke	5.709 cubic feet.
Space in clearance and steam passages at one end of low pressure cylinder	.17 cubic feet.
Ratio of clearance to piston displacement	2.96 per cent.
Area of low pressure steam port	12¼ square inches.
Area of low pressure exhaust port	24½ square inches.
Clearance of low pressure piston	
Number of air pumps	4.
Diameter of each air pump	10 inches.
Net area of the valves of each air pump	47 square inches.
Stroke of the air pumps	12 inches.
Aggregate piston displacement of the air pumps of one engine for one revolution	2.18 cubic feet.
Number of feed pumps	4.
Diameter of the feed pumps	2¼ inches.
Stroke of the feed pumps	6 inches.
Aggregate displacement of the feed pump pistons for one revolution of the engine	0.055 cubic feet.
Diameter of the condenser	24½ inches.
Length of the condenser	8 feet 4 inches.
Number of tubes in the condenser	128.
External diameter of the condenser tubes	1 inch.
Length of the condenser tubes	7 feet 5 inches.
Aggregate condensing surface	248.5 square feet.
Square feet of condensing surface per indicated horse-power	3.17
Length of the connecting rods between centers	64 inches.
Area of the neck of each connecting rod	5 square inches.
Diameter of the cross head journals, H. P	2¼ inches, L. P., 2½ inches.
Length of the cross head journals, H. P	2½ inches, L. P., 3 inches.
Diameter of the crank pin journals	3 inches.

Length of the crank pin journals	3¾ inches.
Number of main journals on each shaft	2.
Diameter of the main journals	6 inches.
Length of the main journals	11 inches.
Diameter of the beam-center journals	5 inches.
Length of the beam-center journals	9 inches.
Diameter of the fly wheel	7 feet 11 inches.
Area of cross section of the rim of the fly wheel	48 square inches.
Weight of the fly wheel	7200 lbs.
Diameter of the pump plungers	14½ inches.
Diameter of the pump rods	2½ inches.
Net area of each pump plunger	162.68 square inches.
Net displacement of each plunger, per stroke, in gallons	15.493
Aggregate displacement of the two pump plungers for one revolution of the engine	14318.62 cubic inches.
Number of suction valves in each end of each pump	35.
Diameter of opening of each valve	$1\frac{5}{16}$ inches.
Aggregate area of opening of the suction valves for each end of each pump	47.25 square inches.
Ratio of area of plunger to area through the suction valves	3.44 to 1.
Number of discharge valves in each end of each pump	34.
Diameter of opening of each valve	$1\frac{5}{16}$ inches.
Aggregate area of opening of the discharge valves of each end of each pump	45.90 square inches.
Ratio of area of plunger to area through the discharge valves	3.54 to 1.
Number of air chambers on each engine	2.
Diameter of the air chambers	2 feet.
Height of the air chambers	4 feet 6 inches.

THE TERMS OF THE CONTRACT.

According to the terms of the contract, the duty test was to continue for 24 hours consecutively, during which time it was to work smoothly and steadily, without heating or vibration, at the contract rate of delivery, which is 2,500,000 gallons of water against a head of 140 feet, at a piston speed not exceeding 110 feet per minute. The contract requires a duty of 90,000,000 foot-pounds for 100 pounds of coal, "based on an evaporation of ten to one." The contract also provides that the computation for capacity shall be made from the plunger displacement, mak-

ing due allowance for slip as the Engineer Commissioner shall determine.

The contract further provides that the duty shall be computed by the formula

$$\text{Duty} = \frac{\text{P N H} \times 100}{\text{W}} \qquad [1.]$$

in which P = pounds of water delivered per stroke.

N = total number of strokes.

H = total fluid resistance, including static and frictional heads, and allowing one pound for friction through the pumps and passages.

W = number of pounds of coal burnt during the test, not computing the coal used for raising steam.

By the term "an evaporation of ten to one" I understand, in this case, a vaporization of ten pounds of water for each and every pound of coal, from the temperature of the feed water, and at the temperature due to the boiler pressure employed.

The amount of water vaporized by a given weight of coal is universally conceded to be the proper measure of the efficiency of a boiler, but in the comparison of boilers, they should have equally good coal, be fed with water of the same temperature and furnish steam at the same pressure. The comparison is properly made by reference to the *equivalent vaporization from and at* 212 *degrees per pound of combustible;* that is, what the vaporization would have been had the coal been without ashes or refuse of any kind; the feed water delivered to the boiler at the boiling point and vaporized under the atmospheric pressure alone.

By reference to the contracts for pumping engines, between E. P. Allis & Co. with the city of Hannibal, Mo., with the city of Alleghany, Pa., and with the city of Decatur, Ill., and Robert Wetherill & Co. with the city of Philadelphia, Pa., and with Cincinnatti, Ohio, I find the stipulation of a vaporization of ten pounds of water per pound of coal, as in this case, and that the contractors understand it to mean, unequivocally, that their duty is to be based on a vaporization of ten pounds of water for each and every pound of coal, from the normal temperature of the feed water.

MANNER OF MAKING THE TESTS.

The scales for weighing the feed water and for weighing the coal were tested by the Sealer of weights and measures, and found to be correct.

The pressure gauges were compared with a certified gauge, and corrections made for them where errors were found. The thermometers agreed with the standard. The indicators, which were borrowed from the Bureau of Steam Engineering of the Navy, were tested and verified.

The indicators used in testing the west engine were of the Thompson make, and those used in testing the east engine—which came later—were of the Tabor make.

The coal was weighed by Clerks detailed from the Commissioner's offices, as it came from the bunker. It was George's Creek Cumberland coal, the kind furnished by the contractor, and was used just as it came; there was no attempt at selecting the better portion.

A pyrometer was placed in the uptake; its error, by comparison with thermometers at 70 and 210 degrees, showed a constant error—in excess—of 241 degrees. The record of this temperature is, therefore, that shown by this instrument, minus 241.

The pressure in the mains was indicated by pressure gauges, which were level with the pump barrels, center to center, and about three feet from the pumps. The pressure in the rising mains was measured by a similar gauge, the distance of which, below the center line of the pump, was accurately ascertained by leveling.

The feed water was weighed in a cask, which emptied into a second cask, from which latter it was drawn by the feed pumps.

The contractors' experts ran the engines and the boilers to suit themselves, without interference on either my part or that of the District employés. The coal was weighed and delivered to the fire room in charges of 250 pounds each, and the firemen were instructed to regulate the fire and the draughts so as to keep a pressure on the boiler gauge of nearly 100 pounds. The ash pit

doors were kept closed nearly all the time, to reduce the rate of combustion. The ashes and the clinkers were weighed dry. As the engine and the boiler continued on their regular duty, the fires were not hauled at the end of the tests. From the weights of ashes thus obtained (about 5 per cent.) it is evident that, in cleaning the fires, the fair proportion had not been withdrawn. The average percentage of ashes for the ten days preceding and succeeding the tests was exactly ten per centum of the coal used. This quantity will, therefore, be used in the calculation for evaporation.

The steam jackets were not trapped, and there was no means of knowing how much live steam was blown through them into the feed water; they discharge into the feed water heater (and into the feed water), the supply being regulated by throttling the steam *before* it reached the jackets. The extraordinary temperature of the feed water would indicate that live steam was freely blown into it.

The head of water against which the engines worked when pumping into the high service mains, was about 57 feet, but when delivering into the stand pipe, about 140 feet. Usually the pumps are not required to pump into the stand pipe for a longer period than two hours daily. The resistance of 140 feet, stipulated in the contract, was obtained by partially closing a stop valve in the main, about 50 feet from the pump house. This valve was regulated by the superintendent of the water works. During the test of the west engine an indicator diagram was taken each hour from each end of each steam cylinder and each water cylinder. During the test of the east engine the diagrams were taken half-hourly. During both tests the pressure in the boilers, the vacuum in the condensers and the pressure in the mains were observed every 15 minutes, averaged and recorded at the end of the hour. The counter, the temperatures of the main, the rising main, the feed water, the uptake and the atmosphere, and the height of the barometer were noted hourly; the quantity of coal and the casks of feed water were charged as they were weighed.

The same boiler was used for both tests; it had been in operation about five days before the first test was begun, and its tubes

had not been swept during that period. After completing the first, and before commencing the second test, the tubes were swept.

At the moment of commencing the tests the counter was noted, the condition of the fire was observed, and the height of the water in the gauge glass on the boiler was marked in the usual way. The tests continued, each precisely 24 hours, uninterruptedly, during which time the steam pressure did not vary more than three pounds, nor the vacuum more than an inch of mercury; the throttles remained wide open, the engines being regulated by the cut-off.

The speed varied between 30.1 and 32.5 revolutions per minute. An undue hammering in the water end of the west engine at the end of the 13th hour was caused, no doubt, by the pressure on the suction side, which had risen to seven pounds; it ceased when that pressure fell. It could have been stopped by partially closing the valve in the rising main. To prevent this hammering, under such conditions, it is the practice of some of the best builders to put springs on the suction valves.

The feed pump of the west engine, soon after commencing the test, stopped working; it was probably full of steam (from the jackets.) The contractors' experts relieved the pressure on the pump by permitting a small quantity of the feed water to escape; the pumps gave no further trouble, except requiring frequent tightening of the plunger packings.

The Westinghouse air-pump was used two minutes, only, during the 24-hour test of the west engine, and not at all during the test of the east engine.

During the test of the west engine the safety valve lifted, and a quantity of steam escaped; it was closed as quickly as possible. No account could be made, nor was credit given for the loss of water or heat by this accident, nor did the contractors' experts claim any.

Passed Assistant Engineers Richard Inch and C. W. Rae, whom the Navy Department detailed for the purposes, stood regular watches with the writer, taking the indicator diagrams, making the observations and recording the data.

"The slip was measured by stopping one engine and putting the normal pressure of the high service mains upon the discharge valves and on one side of the plunger of one of the pumps (which was taken at random); the space on the other side of the plunger was piped to a cask, in which the leakage was weighed. The valves from the rising main to the pump were tightly closed and, being new, were assumed to be tight. In this way all the water which passed through the plunger or through the discharge valves was trapped and weighed. The capacity of the cask was 992 pounds, so the duration of the test was short. A mean of five tests gave a slip of 0.96 per cent. of the plunger displacement, at a mean head of 52.49 feet, and calculated at the plunger displacement at the time of the duty test. The slip will vary as $\sqrt{h}$, so that the mean slip for the west engine will be

$$\frac{\sqrt{143.02} \times 0.96}{\sqrt{52.49}} = 1.58 \text{ per cent.,}$$

and for the east engine

$$\frac{\sqrt{144.03} \times 0.96}{\sqrt{52.49}} = 1.59 \text{ per cent.}$$

To check this measurement of the slip, I had the valves leading to the distributing mains of the high service closed, and the water pumped directly into the stand pipe, a vessel holding 270,919 U. S. gallons, located on a hill about 2,000 feet from the pumps. Two observers were placed on the stand pipe, and, by an electric signal, announced to the observers in the engine house the instant the water began to flow on the floor of the stand pipe, and again when it reached a given height. The observers in the engine house noted the counter at each signal and noted, at intervals of 15 minutes, the pressure indicated by the gauge on the mains. From the data thus obtained the slip appeared to be six per cent. The engine was then stopped, when it was observed that the head diminished from 159.4 to 90 feet in about 15 minutes, due, no doubt, to a leak either in the main or in the valve. There is a check valve which holds the water in the stand pipe; consequently the diminution of pressure was in the pipe alone, which accounts for the rapid diminution. The existence of this

leak of course barred the test for slip. The original measurement was, therefore, accepted and assumed to be correct.

The engines were taken off their regular duty, several days after the duty test, and the pump heads taken off, the suction and discharge valves having been closed. The engines were then started and three complete sets of indicator diagrams taken, to ascertain the power required to work each engine and its concomitants. A jet was used in the condensers to condense the steam; all the air pumps and feed pumps (four of each to each engine,) remained connected. The engines were run at the same speed, as nearly as possible, as on the days of the duty tests. The extraordinary pressure per square inch of piston required to work the engines (about 9 pounds in the high pressure and about 0.8 in the low,) is easily understood when it is remembered that each engine has four air pumps, four feed pumps, six piston rods, two pump rods, twelve valve rods, four steam pistons and two water plungers, besides the beam centers, crankshaft bearings, cross-head guides, etc., to interpose friction.

Capacity of the west engine:

$15.493 \times 4 \times 44585\,(1 - 0.0158) = 2{,}719{,}366$ gallons of water in 24 hours.

Duty of the west engine:

$$\frac{126.64 \times 4 \times 44585 \times (143.02 - 7 + 3.55) \times 100}{4123} = 76{,}453{,}682$$ foot pounds of water lifted for 100 pounds of coal.

Duty of the west engine referred to the contract basis of 10 pounds of water vaporized per pound of coal, and the addition of one pound (2.31 feet) to the head:

$$\frac{126.64 \times 4 \times 44585 \times (143.02 - 7 + 3.55 + 2.31) \times 10 \times 100}{4123 \times 7.681} = 101{,}183{,}587$$ foot pounds per 100 pounds of coal.

Rate of delivery at contract speed of piston of 110 feet per minute:

$15.493 \times (1 - 0.0158) \times 4 \times 30 \times 1440 = 2{,}634{,}890$ gallons in 24 hours.

Capacity of the east engine:

$15.493 \times 4\,(1 - 0.0159) \times 44354 = 2{,}705{,}002$ gallons of water in 24 hours.

Duty of the east engine:

$$\frac{126.71 \times 4 \times 44354 (144.03 - 5 + 3.55)}{3862} = 82{,}998{,}124$$ foot pounds per 100 pounds of coal.

Duty of the east engine referred to the contract basis of 10 pounds of water vaporized per pound of coal, and the addition of one pound (2.31 feet) to the head:

$$\frac{126.71 \times 4 \times 44354 (144.03 - 5 + 3.55 + 2.31) \times 10 \times 100}{3862 \times 8.287} = 101{,}772{,}977$$

foot pounds per 100 pounds of coal.

FULFILMENT OF THE CONTRACT.

The engines worked during the whole of the period of the tests, very steadily, without any heating or jarring, though running above the contract speed. I have no doubt they can be run, continuously, 20 per cent. above that speed without injury.

The *duty* of the west engine exceeded the requirements by 14.25 per cent.; the duty of the east engine exceeded the requirements by 14.85 per cent.

The capacity of the engines at 110 feet piston speed per minute exceeded the requirements by 5.39 per cent. in the west engine and by 5.38 per cent. in the east engine.

REMARKS.

The large amount of heating surface in the boiler, in proportion to the area of the grate, and the low rate of combustion would premise a high economic efficiency of the boiler. The comparative failure of the boiler, in this respect, is due, I have no doubt, to the in-leak of cold air through the brick work which envelopes the boiler. Had the boiler been more efficient the duty of the engine, *ceteris paribus*, would have been greater.

The method adopted by the contractor of measuring the steam jackets is not the best; it would be better to provide proper traps and to keep the boiler pressure in the jackets when they are used. This will have the effect of utilizing the heat in the steam cylinders—where it is most needed—and, when properly trapped, will deliver water and not steam to the feed pumps.

The engines would be improved by substituting hard brass for the cast iron feed pump plungers, which seem to cut out the

packing rapidly; the stuffing boxes on these plungers would be improved if made deeper.

The gearing for adjusting the point of cutting off, which is used to regulate the speed of the engine, is inconvenient and is not easy to manage; a spiral gear could be substituted for the present hand lever at a moderate expense.

The quality of the material, the character of the castings and forgings, and other fitting and finish of the engine are fully equal to that in first-class pumping engines. A table of the data and results is appended.

TABLE CONTAINING THE DATA AND RESULTS OF THE DUTY AND CAPACITY TESTS OF THE HIGH SERVICE PUMPING ENGINES OF WASHINGTON CITY IN THE MONTH OF AUGUST, 1888, BY G. W. BAIRD, PASSED ASSISTANT ENGINEER, U. S. NAVY.

	The West Engine.	*The East Engine.*
1 Date of commencing the test	8 Aug., 1888.	13 Aug., 1888.
TOTAL QUANTITIES.		
2 Duration of the tests, in hours	24.	24.
3 Total number of revolutions made by the engines	44,585.	44,354.
4 Total number of pounds of feed water pumped into the boiler	31,671.	32,007.
5 Total number of pounds of coal consumed	4,123.	3,862.
6 Total number of pounds of ashes	224.	163.
7 Total number of pounds of combustible consumed	3,899.	3,699.
8 Total number of gallons of water lifted	2,719,366.	2,705,002.
MEAN QUANTITIES.		
9 Steam pressure in the boiler in pounds per square inch above the atmosphere	95.81	95.81
10 Fraction completed of the stroke of the H. P. piston, when the steam was cut off	0.2618	0.2768
11 Number of times the steam was expanded	18.01	16 95
12 Height of the barometer in inches of mercury	29.978	30.09
13 Vacuum in the condenser in inches of mercury	25.260	24.80
14 Pressure in the condenser in pounds per square inch, above zero	2.316	2.597
15 Head of water, in feet, in the main	143.02	144.03
16 Head of water, in feet, in the rising main	7.	5.
17 Number of revolutions, per minute, of the engines	30.961	30.801

18	Number of pounds of feed water pumped into the boiler per hour..	1,319.625	1,333.625
19	Number of thermal units imparted to the feed water per hour.....	1,294,565.3	1,319,141.6
20	Height of center line of pump above center of pressure gauge on rising main, in feet.....	3.55	3.55
	MEAN TEMPERATURES.		
21	Temperature of the atmosphere.....	89.12	77.20
22	Temperature of the water in the rising main....	83.30	80.58
23	Temperature of the water in the main	85.50	81.50
24	Temperature due to the steam	334.92	334.92
25	Temperature of the feed water.....	203.58	196.20
26	Temperature of the uptake.....	411.3	326.6
27	Total number of pounds of water lifted.....	22,584,662.	22,480,117.
28	Mean number of pounds of water lifted per hour	941,043.	941,843.
29	Mean number of pounds of water lifted per stroke	126.64	126.71
	STEAM PRESSURES PER INDICATOR.		
30	Steam pressure on the high pressure pistons at the commencement of the stroke, in pounds per square inch above zero.....	108.986	110.388
31	Steam pressure on the high pressure pistons, in pounds per square inch above zero, at the end of the stroke.....	36.945	39.781
32	Mean back pressure against the high pressure pistons during its stroke, in pounds per square inch above zero.....	15.956	17.483
33	Mean pressure per square inch on the high pressure pistons required to work the unloaded engine.....	8.9583	9.6875
34	Indicated pressure on the high pressure pistons, in pounds per square inch.....	53.826	54.380
35	Net pressure on the high pressure piston, in pounds per square inch.....	44.868	44.693
36	Total pressure on the high pressure pistons, in pounds per square inch above zero.....	69.782	71.863
37	Pressure on the low pressure pistons at the commencement of the stroke, in pounds per square inch above zero.....	27.85	27.554
38	Pressure on the low pressure pistons at the ends of the strokes, in pounds per square inch above zero.....	10.667	10.793
39	Mean back pressure against the low pressure pistons during the stroke, in pounds per square inch above zero.....	4.166	3.507

40 Mean pressure per square inch on the low pressure pistons required to work the unloaded engines	0.7375	0.8125
41 Indicated pressure on the low pressure pistons, in pounds per square inch	11.135	12.196
42 Net pressure on the low pressure pistons, in pounds per square inch	10.398	11.384
43 Total pressure on the low pressure pistons, in in pounds per square inch above zero	15.301	15.706
HORSES-POWER.		
44 Indicated horses-power developed in the two high pressure cylinders	41.368	41.580
45 Indicated horses-power developed in the two low pressure cylinders	34.351	37.433
46 Aggregate indicated horses-power developed by the engine	75.719	79.013
47 Net horses-power developed in the two high pressure cylinders	34.485	34.173
48 Net horses-power developed in the two low pressure cylinders	32.043	34.916
49 Aggregate net horses-power developed by the engine	66.528	69.089
50 Total horses-power developed in the high pressure cylinders	53.639	54.934
51 Total horses-power developed in the two low pressure cylinders	47.223	48.205
52 Aggregate total horses-power developed by the engine	100.862	103.139
CONSUMPTION OF FEED WATER.		
53 Number of pounds of feed water consumed per hour per indicated horse-power developed by the engine	17.569	16.878
54 Number of pounds of feed water consumed per hour per net horse-power developed by the engine	19.835	19.303
55 Number of pounds of feed water consumed per hour per total horse-power developed by the engine	13.083	12.930
56 Thermal units consumed per hour per indicated horse-power developed by the engine	17,096.967	16,695.247
57 Thermal units consumed per hour per net horse-power developed by the engine	19,459.83	19,093.36
58 Thermal units consumed per hour per total horses-power developed by the engine	12,835.015	12,789.93

STEAM ACCOUNTED FOR BY THE INDICATOR.

59 Pounds of steam present per hour in the high pressure cylinders at the end of the stroke of the pistons, calculated from the pressures there	1,089.4507	1,186.8906
60 Pounds of steam present per hour in the two low pressure cylinders, at the end of the strokes of the pistons, calculated from the pressures there	1,221.5554	1,235.1864
61 Difference, in pounds, per hour between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicators, in the two high pressure cylinders at the ends of the strokes of the pistons	230.174	146.734
62 Difference, in per cent., of the weight of the water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the high pressure cylinders at the end of the strokes of the pistons	17.442	11.002
63 Difference, in pounds per hour, between the weight of water vaporized in the boiler and the weight of steam accounted for by the indicators, in the two low pressure cylinders at the ends of the strokes of the pistons	98.070	98.439
64 Difference, in per cent., of the weight of the water vaporized in the boiler, between that weight and the weight of steam accounted for by the indicator in the low pressure cylinders at the ends of the strokes of the pistons	7.431	7.381
65 Per cent. which the aggregate indicated horses-power are of the aggregate total horses-power developed by the engine	75.071	76.608
66 Per cent. which the aggregate net horses-power are of the aggregate total horses-power developed by the engine	65.959	66.986
VAPORIZATION.		
67 Mean number of pounds of water vaporized per hour	1,319.625	1,333.625
68 Total number of pounds of water that would have been vaporized in the boiler had the feed water been supplied at a temperature of 100 degrees Fahrenheit and vaporized under the atmospheric pressure	29,137.32	30,246.61

6.8906

5.1864

6.734

1.002

8.439

7.381

6.608

6.986

3.625

6.61

Fig. 1.

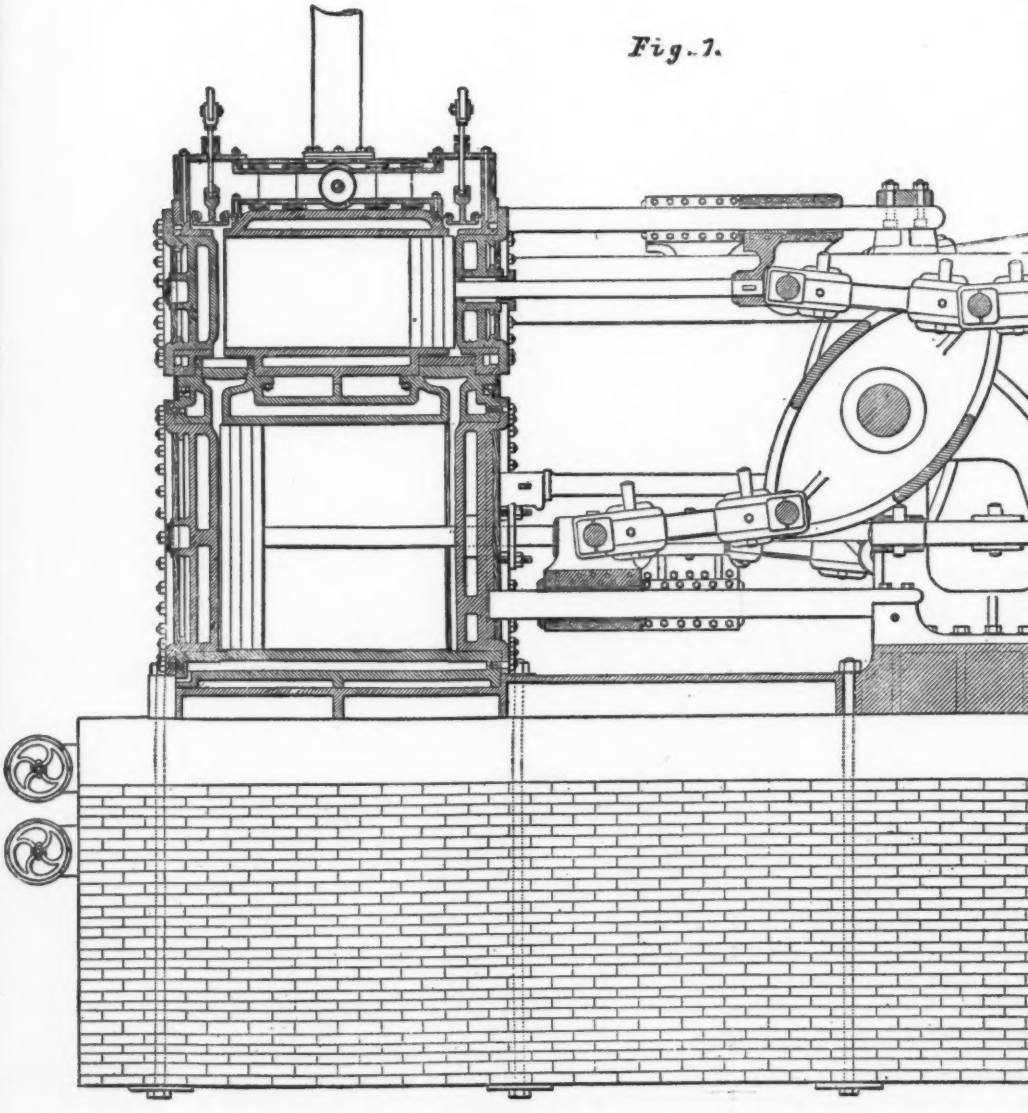
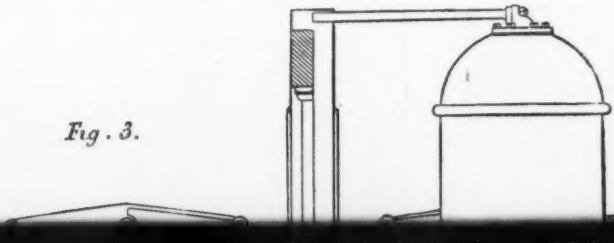


Fig. 3.



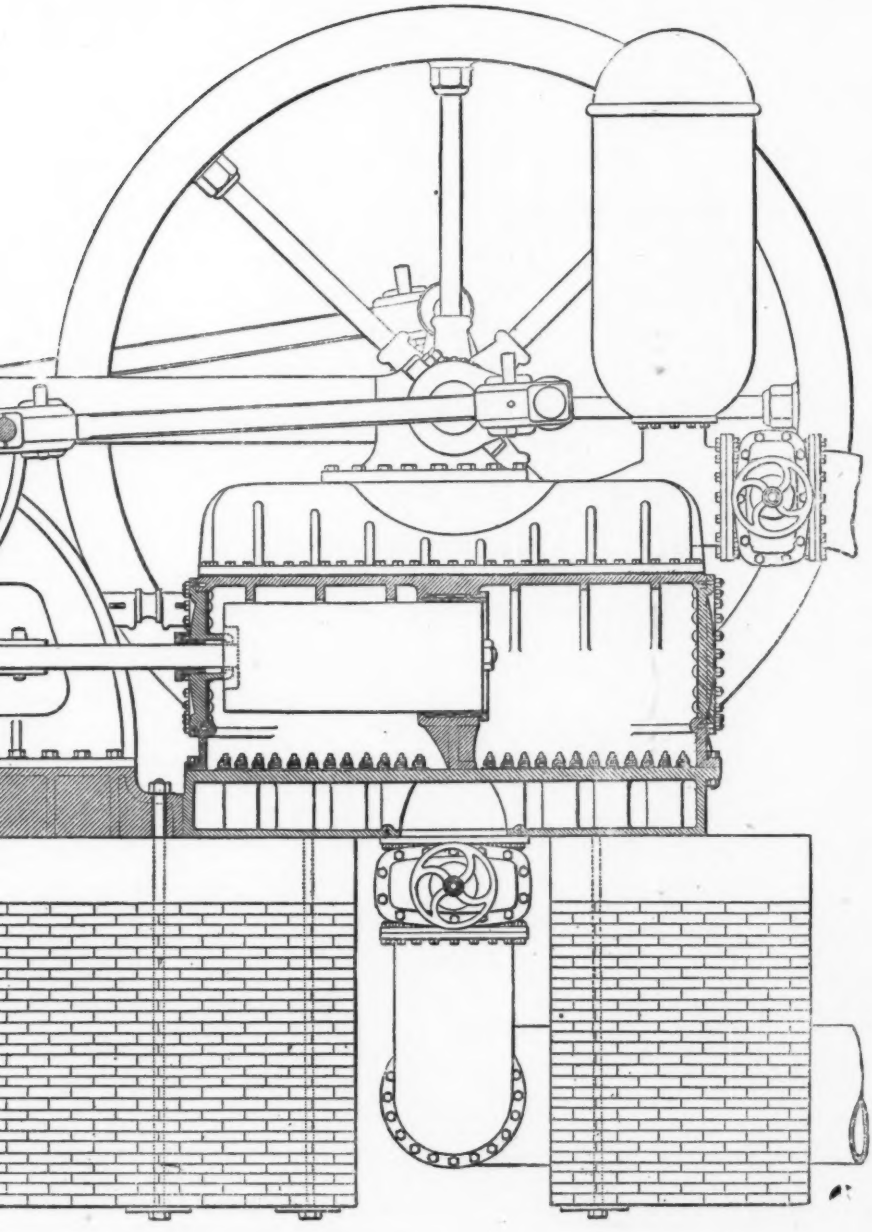


Fig. 2

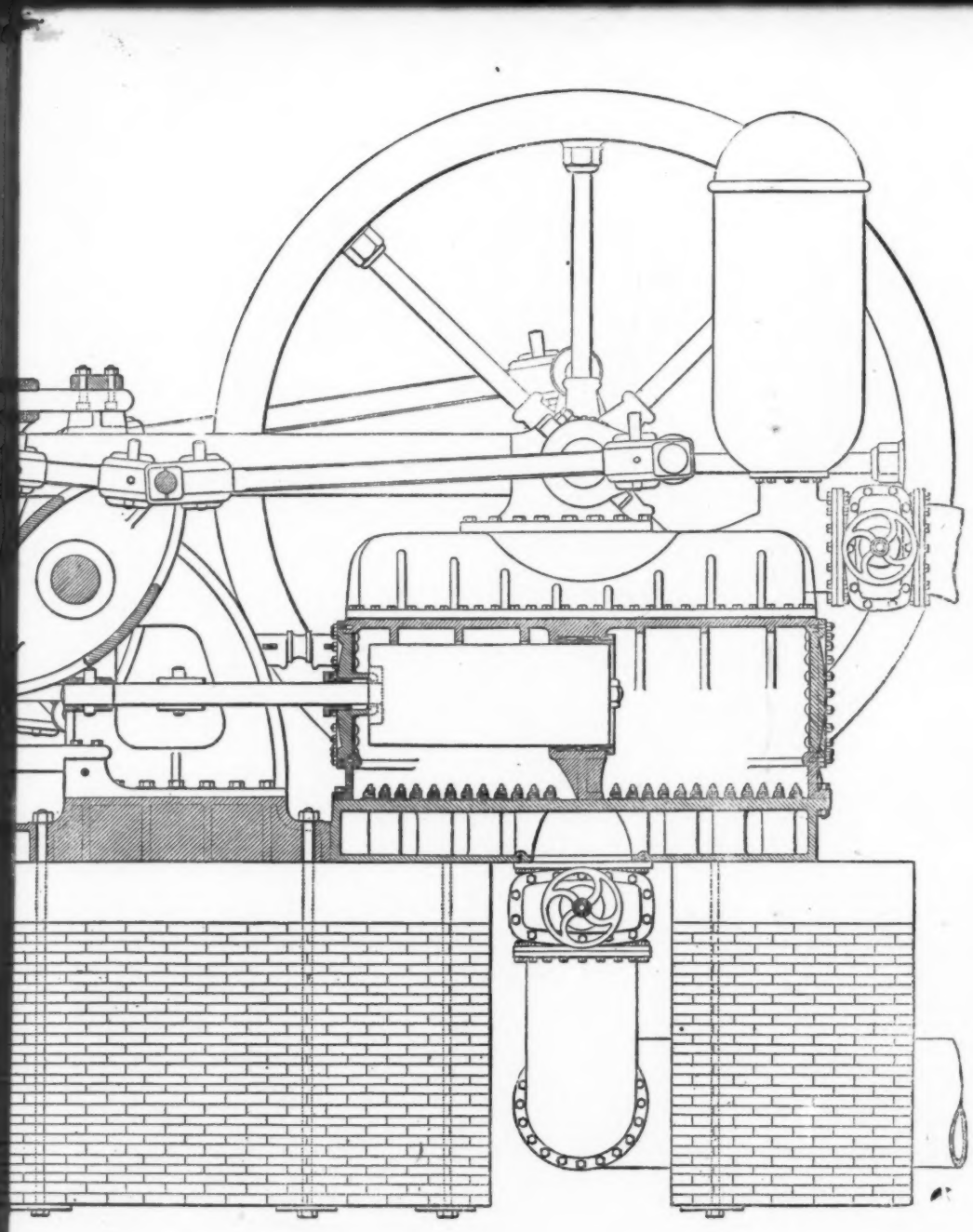
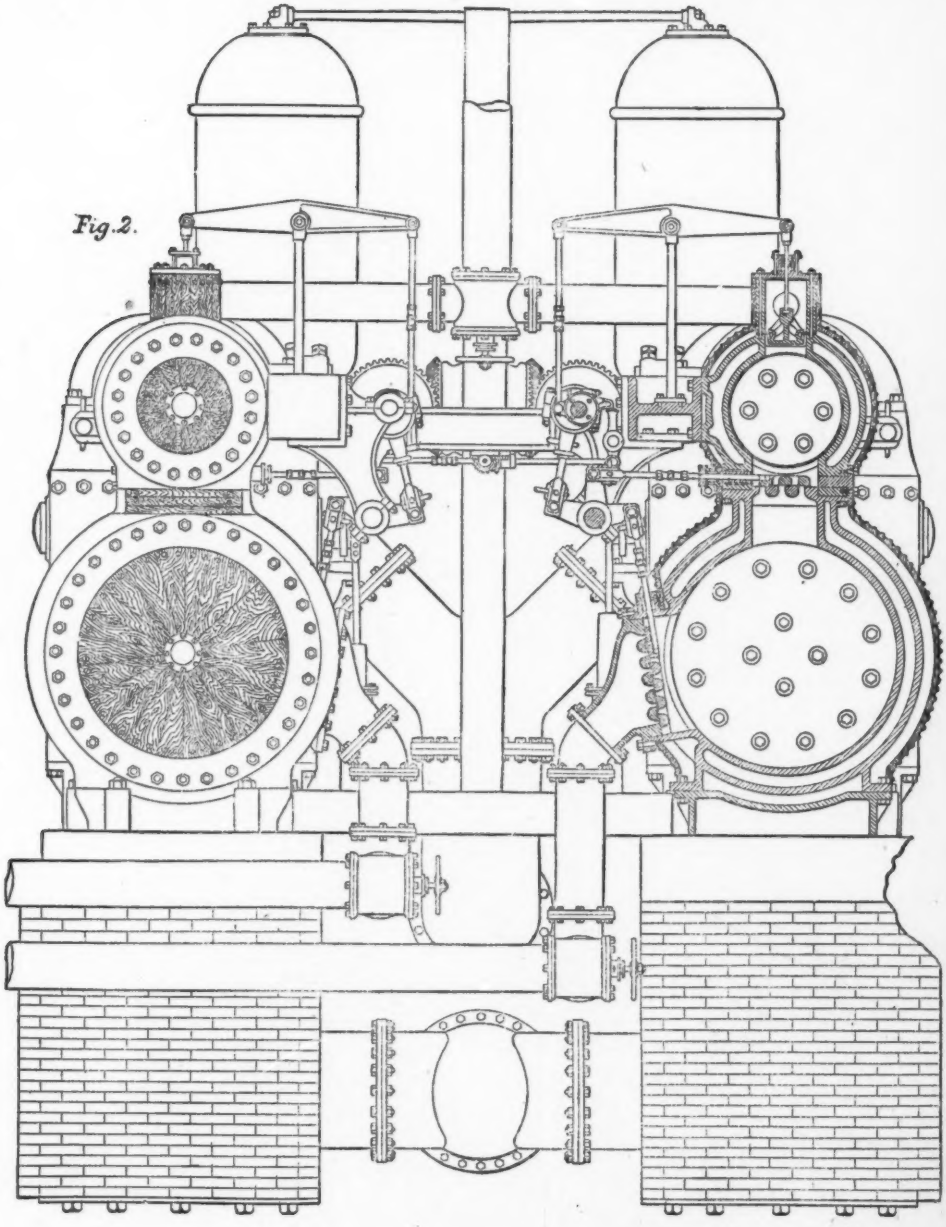


Fig. 2.



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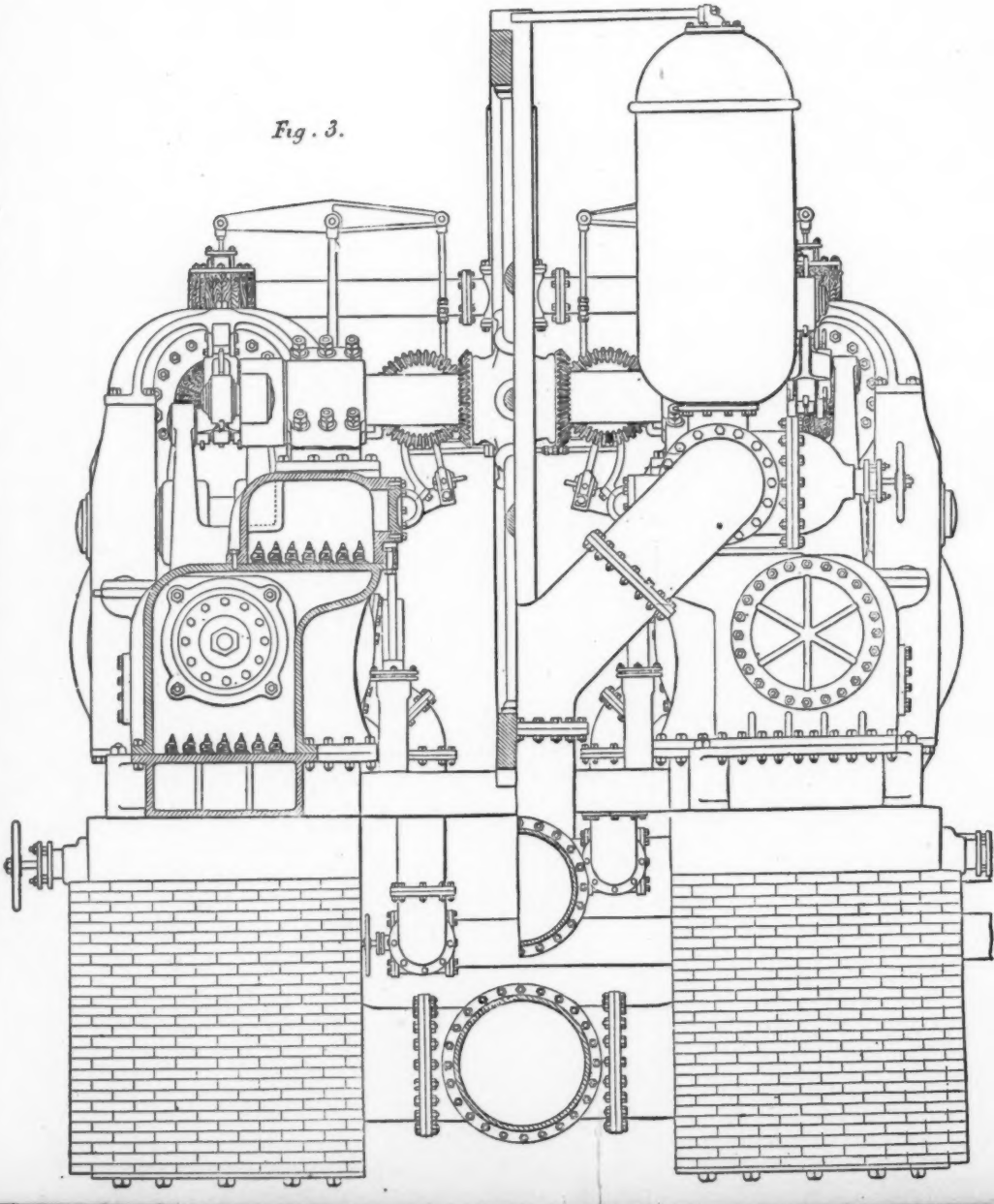

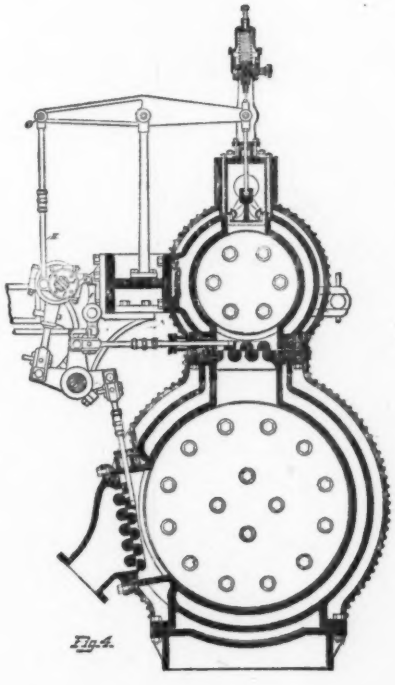

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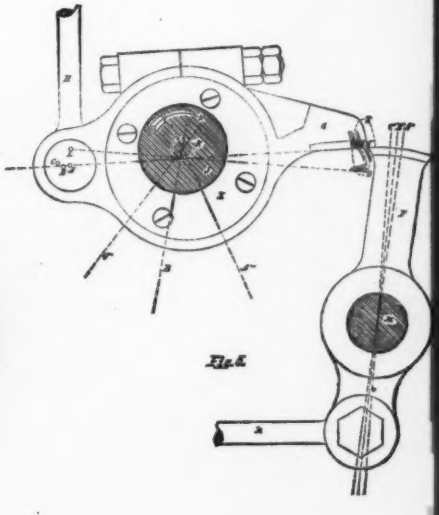

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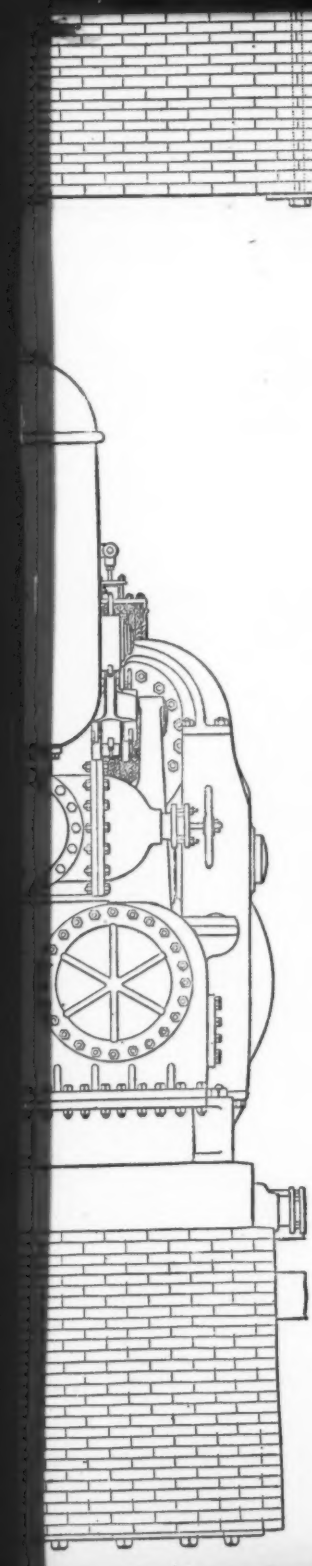


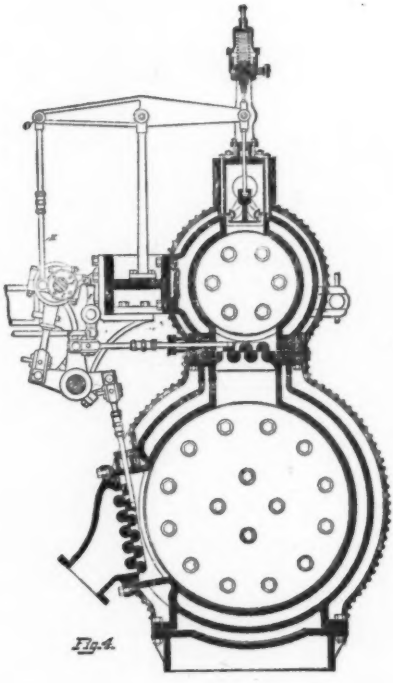
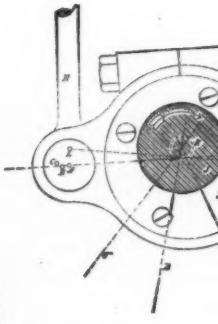
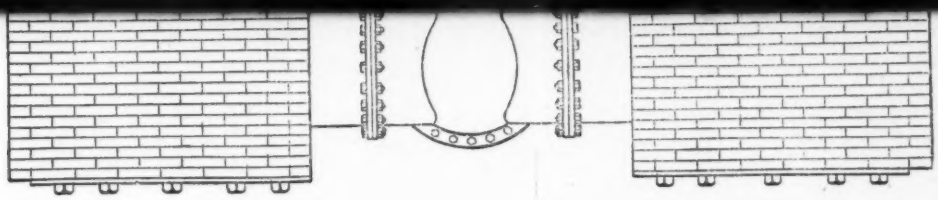


Fig.4.





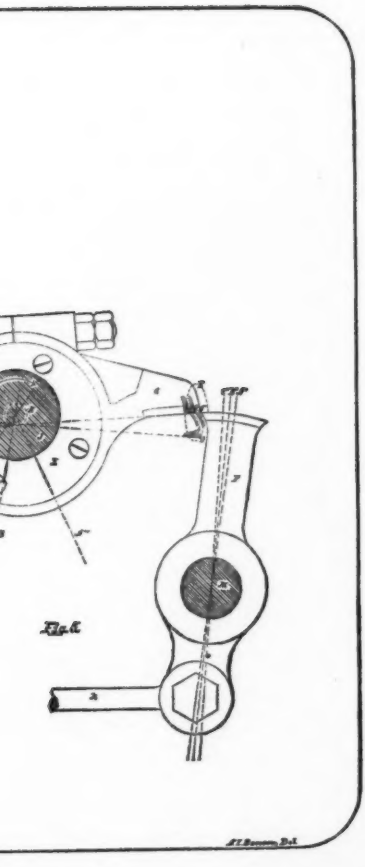
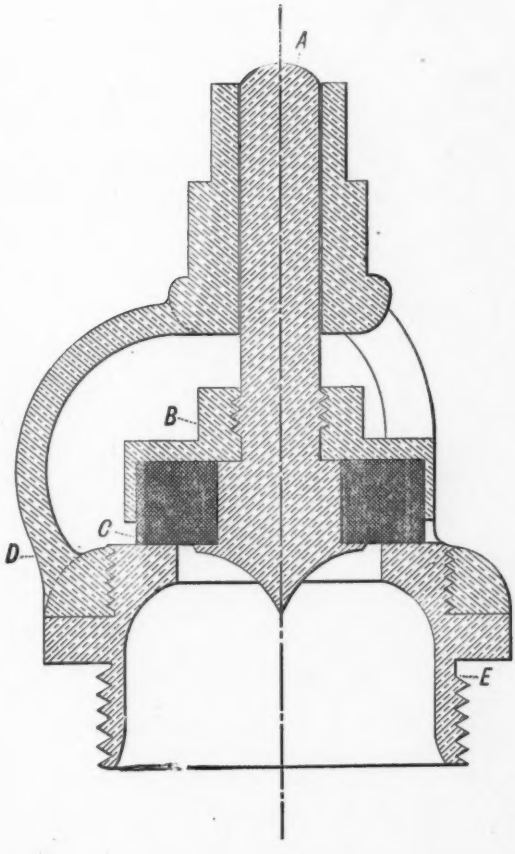


Fig. 6.


A
B
C
D
E

69 Total number of pounds of water that would have been vaporized in the boiler had the feed water been supplied at 212 degrees and vaporized under the atmospheric pressure...	33,222.879	33,767.385
70 Number of pounds of water vaporized from 100 degrees by one pound of coal.....	7.667	7.831
71 Number of pounds of water vaporized from 212 degrees by one pound of coal.....	8.057	8.743
72 Number of pounds of water vaporized from 100 degrees by one pound of combustible.....	7.852	8.702
73 Number of pounds of water vaporized from 212 degrees by one pound of combustible.....	8.953	9.675
74 Number of pounds of water vaporized from the normal temperature of the feed for one pound of coal.....	7.681	8.287
RATE OF COMBUSTION.		
75 Number of pounds of coal consumed per hour,	171.791	160.916
76 Number of pounds of combustible consumed per hour.....	154.875	144.812
77 Number of pounds of coal consumed per square foot of grate per hour.....	5.679	5.319
78 Number of pounds of combustible consumed per square foot of grate per hour.....	5.119	4.708
79 Number of pounds of coal consumed per square foot of heating surface per hour.....	0.209	0.196
80 Number of pounds of combustible consumed per square foot of heating surface per hour,	0.188	0.176

III.

SOME PROBLEMS IN PROPULSION.

By Assistant Engineer W. D. Weaver, U. S. Navy.

The following investigation was suggested after a fruitless search for information on some of the points involved, and although the equations deduced depend upon the ratio between the indicated and effective thrust, of which there is only an approximate knowledge, their use may still prove of value in some cases.

1.°—Suppose that when a vessel is going ahead the engine be instantly reversed:—In what time will the vessel lose her motion ahead, and through what distance will she go before doing so?

The amount of force required at any instant to retard a moving body is equal to the product of its mass by the rate of retardation at that instant; in the case of a ship, this retarding force is the sum of the ship's resistance proper at the given instant and the pull of the screw. We thus have the equation

$$Mr = P + R \tag{1}$$

Assuming the resistance of a ship to be proportional to the square of the velocity, and substituting for the retarding rate and M their respective values $\frac{dv}{dt}$ and $\frac{w}{g}$,

(1) becomes $$\frac{wdv}{gdt} = P + kv^2 \tag{2}$$

or $$dt = \frac{w}{g} \times \frac{dv}{P + kv^2} \tag{3}$$

Substituting in (3) for dt its value from the equation, $\frac{ds}{dt} = v$,

we have $$ds = \frac{w}{g} \times \frac{vdv}{P + kv^2} \tag{4}$$

Integrating (3) and (4) between the limits v_1 and v_2, where v_1 is the greater velocity,

$$t' = \int_{v_2}^{v_1} \frac{dv}{P + kv^2} = \frac{w}{g\sqrt{Pk}} \left(\tan^{-1} \sqrt{\frac{k}{P}} v_1 - \tan^{-1} \sqrt{\frac{k}{P}} v_2 \right) \tag{a}$$

$$s' = \int_{v_2}^{v_1} \frac{vdv}{P + kv^2} = \frac{w}{2gk} \log_e \left(\frac{P + kv_1^2}{P + kv_2^2} \right) \tag{b}$$

in which the units are feet, pounds and seconds.

2.°—The time and distance required to attain full headway from rest.

In this case the resistance of the ship acts against the pull of the propeller, and, following the same reasoning as in Case 1°, we have

$$t'' = \frac{w}{g}\int_{v_2}^{v_1} \frac{dv}{P - kv^2} = \frac{w}{2g\sqrt{Pk}} \log_e \left(\frac{P + \sqrt{Pk}(v_1 - v_2) - kv_1v_2}{P - \sqrt{Pk}(v_1 - v_2) - kv_1v_2} \right) \tag{c}$$

$$s'' = \frac{w}{g}\int_{v_2}^{v_1} \frac{v\,dv}{P - kv^2} = \frac{w}{gk} \log_e \sqrt{\frac{P - kv_2^2}{P - kv_1^2}} = \frac{w}{2gk} \log e \left(\frac{P - kv_2^2}{P - kv_1^2} \right) \tag{d}$$

It is now only necessary to determine the constants to arrive at a solution.

$g = 32.2$

$w = 2240\ D$, where $D =$ displacement in tons.

To Determine k. —

Let $U =$ indicated horse-power at any instant, taken from a speed curve, and

" $v =$ corresponding speed in knots at that instant;

" $f_1 =$ a constant expressing the ratio between the actual and indicated resistance.

Assuming the indicated horse-power to vary as kv^3, we have for the indicated resistance, $kv^2 = \frac{U}{v}$, and the value of the constant k is therefore $\frac{U}{v^3}$. This law is only approximately true, and, as a consequence, to obtain the most accurate result the variation of k should be taken into account; this can be done by finding the values of our final equations (*A*), (*B*), &c., for different intervals by substituting different values of v_1 and v_2 and the value of U corresponding to v_1 taken from a speed curve; by

this method of step-by-step integration, the varying value of k is considered, and the sum of the successive results thus obtained gives the total time or distance, as the case may be.

As the left-hand member of equation (1) is in actual quantities, the right hand member must also be so expressed; multiplying by f_1 to thus reduce it, and substituting for v and U their values in foot-pound-second units, we have

$$k = \frac{550 f_1 U \times (3600)^3}{v^3 \times (6086)^3} \tag{5}$$

To Determine P. —

Let A = indicated horse-power when about to back, and

" a = corresponding effective speed of screw in knots per hour;

" f_2 = a constant expressing the ratio between the actual and indicated pull or thrust.

When a ship is going ahead at a given speed, the indicated thrust is expressed by the quotient of the indicated horse-power at that speed by the speed; if now the engine is instantly reversed and no change made in the admittance of steam to the cylinders, owing to the increased pressure on the blades of the screw the same number of revolutions cannot be made, and the horse-power is proportionately decreased; but, the mean pressure in the cylinders still being the same, the pull backwards, which depends only on this pressure, remains the same as the thrust ahead. In other words, while the horse-power, the numerator of our ratio expressing thrust, is reduced by a certain percentage, the speed, or the denominator, is reduced in the same proportion, the back of the blade being for the present considered as efficient as the face, and the pull remains the same as the original thrust.

The indicated pull is, therefore, $\frac{A}{a}$; multiplying by f_2 for the same reason given for using f_1, and reducing to foot-pound-second units, we have

$$P = \frac{550 f_2 A \times 3600}{6086 a} \tag{6}$$

Substituting the values thus found in (*a*) and (*b*), and reducing to common logarithms and angular measure, we have finally

$$t' = .00631D\sqrt{\frac{av_1^3}{f_1f_2AU}}\left(\tan^{-1}\sqrt{\frac{f_1Ua}{f_2Av_1}} - \tan^{-1}\sqrt{\frac{f_1Uav_2^2}{f_2Av_1^3}}\right) \quad (A)$$

$$s' = \frac{.70355Dv_1^3}{f_1U}\log\left(\frac{f_2Av_1^3 + f_1Uav_1^2}{f_2Av_1^3 + f_1Uav_2^2}\right). \quad (B)$$

A and *a* are not necessarily the values of the horse-power and speed at the instant of backing, but may have any (corresponding) values.

Proceeding with (*c*) and (*d*) in the same manner, but using f_3 instead of f_2, we have

$$t'' = .4162D\sqrt{\frac{av_1^3}{f_1f_2AU}}$$

$$\log\left(\frac{f_3Av_1^2 + \sqrt{f_1f_3av_1AU}(v_1 - v_2) - f_1Uav_2}{f_3Av_1^2 - \sqrt{f_1f_3av_1AU}(v_1 - v_2) - f_1Uav_2}\right) \quad (C)$$

$$s'' = \frac{.70355Dv_1^3}{f_1U}\log\left(\frac{f_3Av_1^3 - f_1Uav_2^2}{f_3Av_1^3 - f_1Uav_1^2}\right) \quad (D)$$

In these final general equations, the values of t and s are in seconds and feet, and the values of D, the speeds, angles and horse-powers are in tons, knots, angular measure and horse-power of 33,000 foot pounds per minute, respectively.

The Numerical Values of f_1 f_2.—

Froude, from towing experiments with a bluff ship, found the ratio of the actual to the indicated resistance to be about .4, while Dupuy de Lôme, experimenting with a finer ship, arrived at a value of .6; in the absence of further data, the mean of these quantities will be taken, which reduced to the notation here employed, gives .45 for the value of f_1.

As the backs of the propeller blades are less efficient than the faces, causing a lateral disturbance of water with consequent loss of force, we will assume our ratio to be diminished to .4 by this cause; and as in losing headway a column of water, the propeller race, strikes the hull with constantly increasing velocity, thus diminishing the effect of pull, we will assume that at the mo-

ment when the vessel is brought to rest the value is further reduced to .35, giving a mean value of .375 for f_2.

Equations C and D could be applied to the case of a ship gaining sternway were it possible to assign values to f_1 and f_3, but, in the absence of data in regard to the increase of the resistance of a ship backing, and the effect of the constantly increasing velocity of the propeller race striking the hull, this cannot at present be done.

The above equations were applied to the "Lepanto" and to a French torpedo boat of 46.1 tons; very full data of the progressive speed trials of the latter are given by M. de Bussy, in the Transactions of the Institution of Naval Architects for 1887, and copious data of the speed trials of the "Lepanto" are in the same publication for 1888.

With these data, speed curves were laid down, and from them the different quantities taken off and applied in the equations, which were solved for progressive intervals to take into account the varying value of k; in backing, the value of f_2 was assumed to vary from .4 to .35 inversely with the speed, while in going ahead f_3 was given the same value as of f_1, .45. The following tables give the results:

LEPANTO

Displacement = 14,860 tons.

Indicated horse-power at 18 knots = 15,040.

Backing.			Ahead.		
Knots.	Time.	Distance.	Knots.	Time.	Distance.
	Sec's.	*Feet.*		*Sec's.*	*Feet.*
18 to 15	24.80	780.8	0 to 3	44.06	112.6
15 " 12	29.95	681.8	3 " 6	46.40	354.1
12 " 9	36.10	648.2	6 " 9	51.45	656
9 " 6	42.97	542.2	9 " 12	63.09	1,132
6 " 3	47.18	364.4	12 " 15	97.75	2,250
3 " 0	52.14	130.1	15 " 17.8	357.70	10,096
Sums....	233.14	3,147.5	Sums.....	660.45	14,600.7

FRENCH TORPEDO BOAT.

Displacement = 46.1 tons.
Indicated horse-power at 19.3 knots = 443.65

Backing.			Ahead.		
Knots.	Time.	Distance.	Knots.	Time.	Distance.
	Sec's	*Feet.*		*Sec's.*	*Feet.*
19.3 to 18	1.151	36.20	0 to 5	8.925	35.62
18 " 15	2.964	82.84	5 " 9	7.389	88.15
15 " 12	3.622	82.51	9 " 12	6.654	118.10
12 " 9	4.328	76.12	12 " 15	9.257	213.10
9 " 5	6.751	79.42	15 " 18	20.530	608.30
5 " 0	9.647	40.43	18 " 19	27.180	855.20
Sums.....	28.463	397.51	Sums.....	79.305	1,918.47

The equations in going ahead were not calculated to the ultimate speed owing to the great increase in the final values for time and distance; indeed, they become infinite for the last tenth of a knot. This is due to the fact that with equal increments of speed the energy required to be imparted to the ship increases, while the force producing this energy, $P - kv^2$, diminishes until in the last increment it becomes infinitesimal.

It will be seen that the rate of retardation in backing with regard to time becomes less at low speeds than at high ones, while with respect to distance it is the reverse. This latter fact is generally recognized, but often ascribed to inefficient action of the screw; as the ratio of the revolutions of the screw immediately after the engines are reversed to its revolutions immediately before, is greater than the slip ratio, it would seem to be turning fast enough to take an efficient hold on the water. When approaching zero the *speed* at which energy is destroyed is slow, and consequently the time large, while at high speed the *amount* of energy to be destroyed is large, and therefore the distance in which to destroy it large.

The relation between the horse-power and the rate of retardation presents a curious anomaly; while the speed of the "Le-

panto" drops from 18 to 15 knots in 24.8 seconds with a mean horse-power of perhaps less than 30 per cent. of its maximum, in going from 3 knots to 0, the time is 52.14 seconds, while in this case the mean horse-power is more than 70 per cent. of the maximum, and the energy destroyed is only $\frac{1}{11}$th of that in the former case. This is due to the fact that the effect of the engines, as far as retarding the vessel is concerned, is a statical rather than a dynamical one, the action of the screw being in a sense comparable to that of a sea anchor or drag astern. The actual energy of the engines is dissipated in the propeller race, and as the velocity of the latter increases with the diminution of the speed of the vessel, there is a consequent and relative increase of horse-power.

In going ahead from rest, at the first instant the entire energy of the engines, more than 75 per cent. of the maximum horse-power, goes entirely into the propeller race; this amount decreases with the speed, until, when the maximum has been attained, the percentage becomes very small, and in most calculations can be neglected. It follows from this that on trial trips, cards taken when the ship has not attained her uniform velocity may be very misleading, and owing to lack of data, it will be difficult to make any correction. Any calculations or comparisons made with such cards are thus valueless; if applied to calculate the horse-power at another number of revolutions by multiplying by the cube of the ratio of revolutions, the power thus obtained will always be too large, and if the card has been taken long before uniform speed is attained, excessively so.

If our equations are integrated between the limits of a and 0, using the mean value, .375, of f_2, the numerical results differ little from those arrived at by step-by-step integration, as the following comparison shows; the same result was found to be true in the case of a vessel of moderate speed and displacement:

	"Lepanto."				Torpedo Boat.			
	Backing.		Ahead.		Backing.		Ahead.	
	t	s	t	s	t	s	t	s
Step-by-step...	233.14	3147.5	660.45	14600.7	28.463	397.51	79.305	1918.47
From a to 0...	234.	3085.	667.	14920.	28.3	399.8	81.47	1968.

This enables us to make a notable simplification in the form of the equations. (A) and (B) become

$$t_1 = \frac{.00631 D a^2}{A\sqrt{f_1 f_2}} \tan^{-1}\sqrt{\frac{f_1}{f_2}} \tag{7}$$

$$s_1 = \frac{.70355 D a^3}{f_1 A} \log\left(\frac{f_1 + f_2}{f_2}\right) \tag{8}$$

in which A and a are the values of the horse-power and speed at the instant of backing. With (C) and (D) the integration is carried to within .2 knots of the final speed, for reasons that have been given before. Their values become, then, between the limits 0 and $a - .2$,

$$t_2 = \frac{.4162 D a^2}{f_1 A} \log (10a - 1) \tag{9}$$

$$s_2 = \frac{.70355 D a^3}{f_1 A} \log\left(\frac{25a^2}{10a - 1}\right) \tag{10}$$

in which A is the horse-power corresponding to the speed a to be attained.

Throughout, the time to reverse the engines and the amount of force required to overcome their inertia have not been considered. On the other hand, it has been assumed that the mean pressure in backing or in gaining speed is the same as at the maximum speed ahead; owing, however, to the slower speeds of the engines, the steam ports will be open a longer time, and the mean pressure will thus be somewhat greater; and an increased boiler pressure, due to less amount of steam used, may still further increase it. In backing from a speed less than the maximum one, the cut-off and throttle may be opened wide, owing to the less amount of steam used, and thus a pressure produced approaching or equal to that of the maximum performance; this should be borne in mind when using the formulæ for this case.

An important application of the formulæ may be made to speed trials. Suppose that a measured distance is to be run over at speeds differing by two knots. After each run, the cut-off and throttle are changed, so as to presumably cause an increase of power sufficient to maintain the additional speed. What time and

distance should be allowed for attaining the increased speed? Owing to the very slow rate at which the last tenth of a knot of speed will be attained, the time and distance to attain one-tenth of a knot less than the desired speed will only be considered. We have, then, from (*C*) and (*D*)

$$t_3 = \frac{.4162Db^2}{f_1 B} \log \left(\frac{20b - 1}{b - 1} \right) \qquad (11)$$

$$s_3 = \frac{.70355Db^3}{f_1 B} \log \left(\frac{400b - 400}{20b - 1} \right) \qquad (12)$$

being the time and speed to go from $(b - 2)$ knots to $(b - .1)$ knots under the conditions assumed, B being the indicated horse-power corresponding to b knots.

These formulæ, applied to the "Lepanto" and torpedo boat, give the following results:—

"Lepanto."			Torpedo Boat.		
Speeds.	t_3	s_3	Speeds.	t_3	s_3
8 to 9.9	770	12170	9 to 10.9	93.41	1634
10 " 11.9	636	12110	11 " 12.9	77.52	1619
12 " 13.9	533	12050	13 " 14.9	63.52	1547
14 " 15.9	445	11560	15 " 16.9	52.84	1462
16 " 17.9	392	11250	17 to 18.9	47.70	1480

Assuming for the constant f_1 and f_2, the values herein used, we finally arrive at the following approximate formulæ, which will probably be found sufficiently accurate for ordinary use:—

Backing. (From a to 0.)

$$t_1 = \frac{.731Da^2}{A} \qquad (13)$$

$$s_1 = \frac{.5354Da^3}{A}$$

Ahead. (From 0 to $a - .2$)

$$t_2 = \frac{2.1Da^2}{A} \qquad (14)$$

$$s_2 = \frac{2.6Da^3}{A}$$

in which A and a have the same values in (13) as in (7) and (8), and in (14) as in (9) and (10).

If formulæ (7), (8), (9) and (10) are applied to the "Etruria," the following results are obtained:—

(Displacement = 9680 tons. Horsepower at 20.18 knots = 14321.)

Backing. (From 20.18 to 0.)	Ahead. (From 0 to 19.8.)
$t_1 = 201.2$	$t_2 = 586.2$
$s_1 = 2974$	$s_2 = 14807$

In the Transactions of the Institution of Naval Architects for 1887 an account is given of the trials of the Russian torpedo vessel "Wiborg", during which the values of t_1 and s_1 were determined by experiment. As it is uncertain on which of the two trials these data were obtained, in the following comparison the calculation has been made for both trials:

Displacement.	Horse-power.	Speed.
138	1303	19.96
131	1405	20.6

From formulæ (13) $t_1 = \begin{cases} 30.84 \\ 28.93 \end{cases}$ $s_1 = \begin{cases} 450.9 \\ 436.4 \end{cases}$

By experiment, $t_1 = 30$ $s_1 = 438.$

IV.

NOTES ON COALS OF THE PACIFIC COAST.

By Passed Assistant Engineer C. R. Roelker, U. S. N.

The following notes are based on observations made during the present cruise of the U. S. F. C. Steamer "Albatross". The steaming efficiency and other characteristics of these coals were determined by actual use at sea on board of this vessel.

The "Albatross" has two main boilers of 12 feet external diameter and 10 feet 3 inches length. Each boiler has three furnaces 36 inches diameter and 6 feet 6 inches length of grate. Area

of grate in each boiler 58½ square feet. Each boiler contains 398 tubes, 3 inches external diameter and 7 feet 9 inches long. Total water heating surface of each boiler 1,424 square feet; ratio of heating to grate surface 24.17 to 1.

CHILEAN COAL.

At Lota, Chile, we received in March, 1888, from "La Compañia Esplotadore de Lota y Coronel" 170 tons of native coal from the Alberto mine. The company uses this coal in its extensive copper smelting works, and most steamers touching at southern Chilean ports use this coal, or a similar coal mined at Coronel on the opposite side of the bay. The price charged for this coal delivered alongside the vessel in lighters was five or six dollars Chilean per ton of 1,000 kilos, for regular consumers, equivalent to $2.60 or $3.10 in our money per ton of 2,240 pounds. Stray customers have to pay a somewhat higher price. For putting the coal on board and stowing it in the bunkers contracts can be made at the rate of twenty cents Chilean per ton.

The coal is not at all prepossessing in appearance. Most of it looks dull, with a peculiar yellowish-grey tint at the planes of cleavage; it breaks up easily in rhomboidal particles. Mixed with it are some more compact lumps, presenting irregular fractures of a deep black, very lustrous. Its bulk is about 11 per cent. greater than that of ordinary anthracite, 46.4 cubic feet being required to stow a ton. It ignites easily and burns rapidly with large flame. It does not cake, but breaks up into small particles in the furnace, which run through the grate unless the spaces between the bars are very narrow. The contents of the ash-pan should always be put back into the furnace.

Large clinkers are formed which spread over the whole grate and form the main part of the refuse, which amounted to 7¾ per cent. of the coal burnt. Much soot is formed, and some trouble was experienced from its ignition in the chimney. Most steamers find it necessary to reduce the calorimeter over the bridge walls when using this coal. After some experiments I made the calorimeter one-seventh of the area of the grate. To limit the consumption of fuel I reduced the grate surface in the boiler to

45 square feet by bricking off the the back of the grates. With these proportions we burned easily in one boiler ten tons per day, or 20.7 pounds per square foot of grate per hour. In fact, to keep the consumption of coal down to this allowance, it was found necessary to clean each furnace of clinkers only once every twelve hours. Better results would probably be obtained with a more rapid combustion and cleaner fires. The steam generating power of this coal was from 75 to 80 per cent. of that of good Welsh coal for equal weights burned in our boilers.

Fire brick of good quality is manufactured by the same company and can be procured at a reasonable price.

SANDY POINT, STRAITS OF MAGELLAN.

The discovery of coal beds was reported some years ago, and great hopes were entertained regarding the prosperity of the place on this account. During our visit at Sandy Point, in January, 1888, I found no evidence that this hope would be realized in the near future. The principal mine had been entirely abandoned; the tramway, built at considerable expense from the town to the mine, was destroyed; the mine shafts had caved in or were filled with water. One shaft, which had been recently opened, was in a workable condition, and the hopeful owner was awaiting the arrival of capital for its further development. So far only a very inferior lignite has been found in this locality.

WELLINGTON COAL.

This coal is, without doubt, the best fuel mined at the present time on the northwestern coast of this continent. In steaming efficiency I have found it only slightly inferior to good Welsh coal. It ignites easily and burns with a long flame. We limited the rate of combustion to 18½ pounds per square foot of grate, but could easily burn more than 20 pounds. It formed about 12½ per cent. of refuse, consisting mainly of a spongy clinker not adhering to the grate bars. It forms considerable soot, and much trouble was experienced from the ignition of the soot in the chimney. A large proportion of the coal received was in lumps

which have considerable consistency. Its bulk is about ten per cent. greater than that of ordinary anthracite coal.

We procured this coal fresh from the mines at Departure Bay, Vancouver Island, where vessels are coaled at the wharves directly from the coal cars and with great rapidity. In ordinary times this coal was sold here for three and a half dollars a ton; but during the present summer and fall, in consequence of the coal famine prevailing on this coast, the price was raised to six dollars a ton, while the coal was sold in San Francisco at fourteen and sixteen dollars a ton. These mines are owned by R. Dunsmuir & Son.

NANAIMO COALS.

The mines which produce these coals lie in close proximity to the Wellington colliery, and the general characteristics of these coals do not differ greatly from those of the Wellington coal. We obtained about one hundred tons of Nanaimo coal at Unalaska, Aleutian Islands, where it had been stored in a closed warehouse. I found it to be 13 per cent. inferior to Wellington coal in steaming efficiency; the amount of refuse produced was nearly 14 per cent. Later in the season we obtained one hundred tons of Nanaimo coal at Kadiak, which had been left exposed to the weather without any covering whatever, and had evidently greatly deteriorated in that damp climate; in addition to this it was quite wet when put into the bunkers. Its steaming efficiency was 18 per cent. less than that of the coal received at Unalaska, and the refuse from this coal amounted to 19 per cent.

SPONTANEOUS COMBUSTION.

All the coals mined at present on the northwestern coast of this continent are of relatively recent formation, and the large percentage of gaseous matter contained in them renders them liable to spontaneous combustion under favorable conditions. Of course each colliery owner strenuously denies that his particular coal has this dangerous property, but the testimony of responsible persons who have used these coals on steamers leaves no doubt in my mind that the Wellington and Nanaimo coals,

as well as those mined near Seattle, W. T., are liable to such accidents. The following case forms part of my own experience:

Five days after we had filled our bunkers with Wellington coal at Departure Bay, smoke and fumes were observed to issue from the bunker. It was found that the covering of the auxiliary steam pipes, which pass through the bunker, was completely charred, and that great heat had been developed in the surrounding coal. No actual ignition of the coal had taken place, but was no doubt prevented only by the timely discovery of the danger. The steam pipes were covered with hair felt and wrapped heavily with manilla rope. The heat of the pipes was probably sufficient to evolve gases from the coal, and by a process of slow combustion of these gases additional heat was generated, causing the charring of the rope and felt covering, and accelerating at the same time the evolution of gases. If this process had been allowed to continue a point would soon have been reached where only a liberal supply of air was wanted to produce active combustion of the coal.

The coal came fresh from the mine, and, as there was a slight rain falling at the time, it was somewhat wet when put into the bunkers. No signs of unusual heat in the bunkers had been observed on any former occasion, either with Wellington or any other coal, although stored in the same manner.

V.

INCREASE OF HORSE-POWER FOR A GIVEN SPEED DUE TO FOUL CONDITION OF SHIP'S BOTTOM.

By Assistant Engineer W. M. McFarland, U. S. Navy.

It is a fact familiar to all acquainted with calculations of power for ship's propulsion that a slight fouling of the wetted surface of the hull will cause a material increase in the resistance, but the writer is not aware of any data having been tabulated to show,

at a glance, exactly how much of an increase of power is rendered necessary. During a recent cruise of the U. S. S. "Vandalia," two opportunities occurred for securing such data, and the subjoined tables are given in the hope that they may prove of some interest and possibly of value.

The "Vandalia" joined the Pacific Squadron in December, 1886, at Valparaiso, and was employed on the west coast of South America until July 21, 1887, when she left for Honolulu. During this time she lay at Panama for about six weeks.

It may not be generally known that marine growth on ships' bottoms is more rapid in the Pacific than almost any other ocean, but inquiry seems to establish the fact; and the harbor of Honolulu is especially noted as being, perhaps, the worst place in the world for rapid fouling of the hull. It may be stated in this connection that the ravages of the *teredo* are very great in this harbor, so much so that the piling for wharves has the part exposed to the water coppered.

The "Vandalia" arrived at Honolulu August 27, 1887, and lay moored in the harbor, except for a few days at target practice outside, until early in March of 1888, a period of about six months. The circumstances were thus very favorable to the marine growth. In March the ship went to Hilo, a distance of about 175 miles, and returned to Honolulu again in two weeks, where she remained until September 19, 1888, making her whole stay in that port about a year.

No effort was made to clean the hull or propeller prior to the trip to Hilo, but the facts shown by Table I. were so marked that before leaving for San Francisco the propeller was thoroughly scraped, and the hull to a distance of about six feet below the water line was cleaned as well as could be done from boats by means of "ship's scrapers" on the end of poles.

During this cleaning, specimens of this growth on the propeller and copper were obtained presenting a most peculiar appearance. Next to the metal was a substance looking somewhat like the outer skin of a black hog, with the bristles removed, which had formed in lumps about the size and shape of a turnip, that is, an

irregular oval with axes about 2½ and 3½ inches, and from 1 to 3 inches thick. The side of this growth which had adhered to the metal looked very much like the "suction side" of a star fish. The outside was not, of itself, very rough, but from it extended almost innumerable tubes of a hard, whitish substance, which is called, among sea-faring people at Honolulu, coral. These tubes were often four inches long, and their external diameter is about one-sixteenth of an inch. An idea can thus be formed of the condition of the ship's bottom, so that the annexed data need not cause great surprise.

The displacement of the vessel was so nearly the same in all the four cases that the difference would have little effect. The wetted surface at the draught in these cases is, by Kirk's analysis, about 11,000 square feet.

The first comparison is between a run of eight hours in the "doldrums" on the way to Honolulu and eight hours on the way to Hilo in March, 1888. This gives us

TABLE I.

	August 17, 1887. Bottom clean, (comparatively.)	March 12, 1888. Bottom foul.
Number of hours	8	8
Wind, points from ahead	2	4
Wind, force	1.5	4
State of sea	Smooth.	Gentle swell.
Speed in knots	7.49	6.24
Revolutions per minute	41.66	40.47
Indicated horse-power	371.2	441.92
Slip of propeller, per cent.	12.98	25.54
Sail set	Fore and aft, but not drawing	Fore and aft.
I. H. P. at 6.24 knots (reduced in ratio of cubes of speeds $= 371.2 \times \frac{(6.24)^3}{(7.49)^3}$)	214.62	441.92

It might be objected that as the conditions of wind and sea are not absolutely identical in the two cases the comparison is not

strictly fair. In Table II. the effort has been made to get the circumstances of wind and sea as nearly the same as possible. The data are for 24 hours, enroute from Panama to Payta, Peru, when the bottom was comparatively clean, and for 13 hours out of Honolulu, en route to San Francisco, in September of 1888.

TABLE II.

	May 21, 1887. Bottom clean.	Sept. 19, 1888. Bottom foul.
Number of hours	24	13
Wind, points from ahead	0 to 3	4
Wind, force	2 to 4	4
State of sea	Moderate.	Moderate.
Speed in knots	7.134	5.27
Revolutions per minute	42.33	36 53
Indicated horse-power	393.4	311.3
Slip of propeller, per cent	18.39	30.29
Sail set	None.	Fore and aft.
I. H. P. at 5.27 knots, (reduced in ratio of cubes of speeds $= 393.4 \times \frac{(527.)^3}{(7.134)^3}$)	158.6	311.3

Some of the data of the "Vandalia's" dimensions are given to assist any examination of the figures in the tables.

Length on water line,	217 feet 5 inches.
Beam,	39 feet 0 inches.
Mean draught (load line),	17 feet 1 inch.
Displacement (load line),	2,084 tons.
Wetted surface at mean steaming draught by Kirk's analysis	11,000 square feet.
Screw, diameter,	15 feet 6 inches.
pitch, mean,	21 feet 0 inches.
developed area,	81.8 square feet.
material,	Composition.

The hull is of wood and is coppered.

Reference Number.	Name of Vessel.	Dimensions of Hull.				Cylinders.			
		Length.	Breadth.	Depth of Hold.	H. P.	Diameters. 1st I. P.	2d I. P.	L. P.	Stroke.
		Feet.	*Feet.*	*Feet.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*
1.	Grace Darling	154	19½	11	*10¾	14	20	28	20
2.	Skeandhu	76	10½	5½	7	9	12½	18	12
3	Bromo	320	36¾	27	23	33	43	63	42
4.	Buenos Aires	410	48	32	32	46½	64½	92	60
5.	Jelunga	410	48	32	.....	.....	.....	.....	.....
6.	Kronprinz Fr. Wilhelm	318	39½	31	21½	30½	43	61	48
7.	Tenasserim	.....	.....	.....	24½	37	49	72	42
8.	City of Venice	379½	38¼	29⅙	30	40	52	70	48
9.	County of York	285	38	25 7/12	20	28½	40	57	42
10.	Myrtle	163	20½	14	12	17	24	34	24
11.	Falls of Inversnaid	321	40⅓	19 7/12	18	26	36	52	39
12.	Suez	.....	.....	.....	22	30	43	62	45
13.	Ballymena	148	18	11½	11¼	16	22½ (2)	22½ (2)	15
14.	St. Oran	58	9½	.....	.....	.....	.....	.....	.....

* 10 is also given, but 10¾ is the more reliable.

Reference Number.	Performances.			Total I. H. P.	
	Speed in knots per hour.	Revol's per minute.	Steam pressure pounds per sq inch.		
1	12½	160	180	360	The I. H. P. d
2	10½+ 10			128.14 100	This power wa From owner's s
7	11¾	60 68		1,700 1,800	On measured m Draught of ves
9	8 5 9.3	 58	 155	780 940 983.6	Highest mean " " Vacuum, 26 in
12		56	152	985.5	Vacuum, 27.5 i

TABLE I.

ke.	Boilers.			Builders of Machinery.		
	Number and Material.	Kind.	Working pressure in pounds per square inch.	Name.	Place.	Whose S of
ies.						
0	1 steel.	2-furnace.	200	Fleming & Ferguson.	Paisley, Scotland.	F. & F.
2	1 "	Return-tube.	180	" "	" "	"
2	2 " (?)	Double-ended.	200	Royal Shipbldg. & Engrg. Co. de Schelde.	Flushing, Holland.	.....
0	3 "	"	†180	Denny & Co.	Dumbarton, Scotland.	Walter
...	....	....	.....	"	" "	"
8	2 "	Double-ended.	170	"	" "	"
2	2 "	"	180	"	" "	"
8	2 "	Single-ended.	145	Jas. Howden & Co.	Glasgow, "	Howden
2	2 main } " 1 aux. }	Return-tube.	164	Barrow Shipbldg. Co.	Barrow-in-Furness.	.....
4	1 "	"	180	Rankin & Blackmore.	Greenock, Scotland.	J. F. &
9	2 main } " 2 aux. }	....	180	" "	" "	"
5	....	....	.....	Central Marine Engrg. Co.	West Hartlepool.	.....
5	1	Herreshoff rect. coil.	220	Herreshoff Mfg. Co.	Bristol, Rhode Island.	Herresh
...	1 "	....	200	Simpson, Strickland & Co.	Dartmouth.	Kingdo

†170 also given.

TABLE II.

	Remarks.
	. derived from one set of indicator cards was 353.27, distributed as follows: II. P. cyl., 81.64; 1st I. P., 95.18; 2d I. P., 76 63; L.
	was derived from indicator cards, the distribution being as follows: H. P. cyl., 26.5; 1st I. P., 28.1; 2d I. P., 31.04; L. P., 42.5. 's statement, the coal consumption being 1.12 pounds of Scotch coal per I. H. P.
	d mile, October 26, 1888. Fully loaded; weather unfavorable. vessel, light.
	n results. Displacement at time of trial, 4,599 tons. Rough sea. " " " " " 2,970 " " inches. The power was derived from indicator cards. Initial pressure in H. P. cyl., 152. I. H. P. on forward crank, 494 7; on a
	.5 inches. The power was derived from indicator cards. I. H. P. on forward crank, 491; on after crank, 494 5.

TABLE I.

SSEL.	Dimensions of Hull.				Cylinders.					
					Diameters.			Stroke.	Number and Material.	
	Length.	Breadth.	Depth of Hold.	H. P.	1st I. P.	2d I. P.	L. P.			
	Feet.	*Feet.*	*Feet.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*		
.....	154	19½	11	*10¼	14	20	28	20	1 steel.	2-fu
.....	76	10½	5½	7	9	12½	18	12	1 "	Ret
.....	320	36¾	27	23	33	43	63	42	2 " (?)	Do
.....	410	48	32	32	46½	64½	92	60	3 "	
.....	410	48	32	.....	.....	.....	.....	.....	....	
ilhelm..	318	39½	31	21½ 24½	30½	43	61	48	2 "	Do
.....	.....	.....	.....	30	37	49	72	42	2 "	
.....	379½	38¼	29⅙		40	52	70	48	2 "	Sin
.....	285	38	25 7/12	20	28½	40	57	42	2 main, 1 aux. "	Re
.....	163	20½	14	12	17	24	34	24	1 "	
id.....	321	40⅓	19 7/12	18	26	36	52	39	2 main, 2 aux. "	
.....	.....	.....	.....	22	30	43	62	45	....	
.....	148	18	11½	11¼	16	22½	22½ (2)	15	1	He
.....	58	9½	.....	.....	.....	.....	.....	.....	1 "	

ut 10¼ is the more reliable.

TABLE II.

Performances.			Total I. H. P.	
n knots per hour.	Revol's per minute.	Steam pressure pounds per sq inch.		
12½	160	180	360	The I. H. P. derived from one set of i
10½+ 10			128.14 100	This power was derived from indicator From owner's statement, the coal consu
11¾	60 68		1,700 1,800	On measured mile, October 26, 1888. Draught of vessel, light.
8 5 9.3	58	155	780 940 983.6	Highest mean results. Displacement " " " " Vacuum, 26 inches. The power was d
	56	152	985.5	Vacuum, 27.5 inches. The power was

Boilers.		Builders of Machinery.		
Kind.	Working pressure in pounds per square inch.	Name.	Place.	Whose System or Patent of Engines.
2-furnace.	200	Fleming & Ferguson.	Paisley, Scotland.	F. & F.
Return-tube.	180	" "	" "	"
Double-ended.	200	Royal Shipbldg. & Engrg. Co. de Schelde.	Flushing, Holland.	....
"	†180	Denny & Co.	Dumbarton, Scotland.	Walter Brock.
....	.....	"	" "	"
Double-ended.	170	"	" "	"
"	180	"	" "	"
Single-ended.	145	Jas. Howden & Co.	Glasgow, "	Howden's.
Return-tube.	164	Barrow Shipbldg. Co.	Barrow-in-Furness.	....
"	180	Rankin & Blackmore.	Greenock, Scotland.	J. F. & Matthew Rankin.
....	180	" "	" "	" " "
....	.....	Central Marine Engrg. Co.	West Hartlepool.	....
Herreshoff rect. coil.	220	Herreshoff Mfg Co.	Bristol, Rhode Island.	Herreshoff.
....	200	Simpson, Strickland & Co.	Dartmouth.	Kingdon.

† 170 also given.

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Remarks.

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nsumption being 1.12 pounds of Scotch coal per I. H. P.

3. Fully loaded; weather unfavorable.

nt at time of trial, 4.599 tons. Rough sea.
" " 2.970 " "
s derived from indicator cards. Initial pressure in H. P. cyl., 152. I. H. P. on forward crank, 494 7; on after crank, 488.9.

was derived from indicator cards. I. H. P. on forward crank, 491; on after crank, 494 5.

VI.

DATA OF SOME QUADRUPLE EXPANSION ENGINES AND THEIR PERFORMANCES.

By Assistant Engineer F. C. Bieg, U. S. Navy.

1. "Grace Darling," steel yacht. Launched, June 8, 1888. Tandem engines; two cranks at 90°; to indicate 400 horse-power, with 200 pounds of steam and on a coal consumption of 400 pounds per hour. The valve gear is of the link type; all valves are slide valves. The boiler is 10 feet in diameter and 9 feet long; two corrugated furnaces $\frac{9}{16}$ inch thick, each 5 feet 4 inches long and 36 inches in external diameter, with a total grate surface of 27 square feet. The tubes are 3 inches in external diameter, No. 8 B. W. G. thick. The total heating surface is 752 square feet. There is one air pump, 12 inches, one circulating pump, 7 inches, and one feed pump, $2\frac{3}{4}$ inches in diameter, all with a stroke of 12 inches and worked by levers from the main engines. The condenser has 390 square feet of cooling surface. The propeller is 3-bladed, solid, 7 feet in diameter and 10 feet mean pitch. The whole of the L. P. and the bottom of the 2d I. P. cylinder is jacketted. The feed water enters the boilers at about 200° F. Darling's patent evaporator and distiller is used to supply the waste of fresh water to boiler. The crank shaft is of wrought-iron, $5\frac{1}{2}$ inches, and propeller shaft, 6 inches in diameter. The three main bearings are each 10 inches long.

2. "Skeandhu," yacht. The four cylinders are in the same horizontal plane; the H. P. and 2d I. P. cylinders being forward and the 1st I. P. and L. P. cylinders aft. A triangular steel casting and short links connect each pair of pistons to its crank. The two cranks are 180° apart. It is claimed that the above combination is equal to one of four separate cylinders with cranks 90° apart. The valve gear is of the link type; all valves are piston valves, having one rod and one chest for each pair of

valves. The air, circulating and feed pumps are worked by levers from the main engines. The propeller has two blades, set aft.

3. "Bromo," steel steamer. The I. H. P. is estimated at 1500. Weir's feed water heater and evaporator are used. The draft is forced by two six feet fans, the air, before entering the furnaces, being heated by the waste gases.

4. "Buenos Aires," s. s. Disconnective tandem engines; two cranks at 90°. The estimated I. H. P. is 4300. The valve gear is of the link type; there is one piston valve for each upper, and two double-ported slide valves, side by side, for each lower cylinder. The H. P. cylinder is the forward upper one and the L. P. cylinder, the after lower one. The upper and lower cylinders have each only one stuffing box between them. There are no pipes or connections to upper cylinders; this arrangement allows these cylinders to be easily removed. The pistons are of stamped steel. The travel of the forward valves is 8 inches and of the after, 13 inches. The connecting rod is 120 inches long. There are four main bearings, each 25 inches long. The crank-pins are each 18 inches in diameter and 19 inches long. The condenser is rectangular, has 1303 tubes of ¾-inch external diameter, and a cooling surface of 4604.8 square feet. There are two air pumps, each 26 inches, two circulating pumps, each 14½ inches, and two feed pumps, each 5½ inches in diameter, all having a stroke of 24 inches, and worked by levers from the main engines. During the trial on the Clyde on Nov. 23, 1888, the ship attained a speed of over 15.5 knots on the measured mile.

5. "Jelunga," steel steamer. Launched, Aug. 22, 1888; gross tonnage, 5,200.

6. "Kronprinz Fr. Wilhelm," s. s., belonging to the N. G. Lloyd of Bremen. Converted from compound engines, 48 inches by 88 inches and 48 inches stroke. The new engines were built at Dumbarton, shipped to Bremerhaven, and there set up in ship by the company's force. The old boilers were single-ended, four in number and working with 60 pounds pressure of steam. The new boilers have 173 square feet of grate, and 4,338 square feet of heating surface. Weir's patent feed-water heater and evap-

orator are used. The circulating water is supplied by a Gwynne's patent centrifugal pump. During the trial on the Weser, on November 7, 1887, the mean speed for six hours, between lights and on the measured mile, was 13.19 knots, the I. H. P. being about 1,700.

7. "Tenasserim," steel steamer. Converted from compound engines, 47 inches by 82 inches and 42 inches stroke. This was the first steamer converted on Brock's system. In this system, the old cylinders and boilers are removed, the old valve gear and frames being utilized for the new engines. All of the new valves are in the casing of the lower cylinders. Weir's feed-water heater and evaporator are used.

8. "City of Venice," s. s., 3,372 gross tonnage. Converted from compound engines, 50 inches by 86 inches and 48 inches stroke. The new three-furnace boilers, 14 feet in diameter and 10 feet 10 inches long, have 80 square feet of grate and 4,006 square feet of heating surface. Ample steam is furnished when engines are developing 1,800 I. H. P. Weir's feed-water heater and evaporator are used. During a recent voyage of this ship, (steaming time, 60 days and 17 hours,) under forced draft, 1,472 tons of coal were consumed, or 24 tons per day. In comparison with this performance, that of a sister ship, fitted with triple expansion engines, is interesting. The same voyage was made, (steaming time 58 days and 20 hours,) under natural draft, the coal consumed being 1,902 tons, or 32.2 tons per day.

9. "County of York," iron steamer, 2,282 gross tonnage. Tandem engines; two cranks at 90°. The valve gear is of the link type; piston valves for H. P. and 1st I. P. cylinders, and Thom's patent slide valve for L. P. cylinder. The condenser is rectangular and contains 1,518 square feet of cooling surface. The air and feed pumps are worked by levers from the main engines. The circulating pump is independent and of the centrifugal type. The main boilers are 12 feet in diameter and 6 feet 6 inches long and have three furnaces each, with a total of 99 square feet of grate and 2,692 square feet of heating surface. The furnaces are Fox's corrugated, ½ inch thick and 2 feet 9 inches in internal diameter. There are 188 tubes in each boiler, 6 feet 7⅛

inches long and 3¼ inches in external diameter. The auxiliary boiler has two furnaces. The propeller is 16 feet in diameter, has a pitch of 18 feet 6 inches and a projected area of 60 square feet.

10. "Myrtle," yacht; tonnage, 318. Disconnective tandem engines, occupying same space as old compound engines; two cranks, counterbalanced. The valve gear is of the link type; piston valves for all cylinders. The condenser has 618 square feet of cooling surface. The air, circulating and feed pumps are single-acting, worked by levers from the L. P. cross-head, the two former being 14 inches and the latter, 3 inches in diameter, the stroke of all being 13½ inches. The boiler is 11 feet 6 inches in diameter by 9 feet 6 inches long, and has 2 Fox's corrugated furnaces, 3 feet 5 inches in internal diameter. There are 52 stay and 115 ordinary tubes, all 3½ inches in external diameter and 7 feet long. The heating surface is approximately 1,395 square feet. The crank shafts are interchangeable. In case of accident to the H. P. cylinder, its stop valve is closed and steam of about 60 lbs. pressure admitted direct to 1st I. P. cylinder. In case the L. P. cylinder is disabled, steam is exhausted from the 2d I. P. cylinder into the main exhaust pipe. A prolongation of the feed pipe passes nine times across the uptake, forming a feed water heater. The propeller is cast-iron, solid, and has four blades set aft. On May 21, 1888, a speed of fully 12 knots was attained, the weather being stormy. On May 25, 1888, during a three hours' trial, the speed was about 11 knots, the consumption of coal being 1.2 pounds of best hand-picked Welsh coal per I. H. P. A trial was made on the same day with the propeller shafting disconnected, but all pumps (including feed and bilge pumps) working. Under these conditions the initial friction of the engines was found to be 8¾ per cent.

11. "Falls of Inversnaid," steel cargo steamer. Tandem engines; two cranks; virtually separate engines. The valve gear is of the link type. The air pump, single acting, and circulating pump, double acting, are worked by levers from the main engines. The main boilers are 11 feet in diameter and 13 feet long. The draft is forced by means of two independent Chandler's

patent silent exhaust engines and fans, blowing into closed ash-pits. These engines (compound) exhaust into the L. P. cylinder casing, making them really triple expansion engines. Weir's feed-water heater and evaporator are used. During the trial on Oct. 18, 1888, with 4040 tons of dead weight on board, the ship made nearly 9¼ knots per hour, consuming at the rate of about 13 tons of Nixon's navigation coal per day. This included the steam used for the steering engine. The firemen were not accustomed to forced draft nor the use of Welsh coal.

12. "Suez," cargo steamer. Tandem engines, converted from compound engines; two cranks. The H. P. and 1st I. P. cylinders are fitted with special piston valves. The yearly wear of these valves is said to be about $\frac{1}{100}$ of the diameter of the valves. On the first run from the North to Lisbon, Portugal, the coal consumption was 34 per cent. less than with the old compound engines, the kind of coal, number of revolutions, speed, and propeller, being the same in both cases.

13. "Ballymena," steel yacht. Launched, Oct. 6, 1888. Five cylinders, side by side, in same fore-and-aft line. The valve gear is of the link type. Independent air and circulating pumps are provided. The condenser is automatic, when the vessel is under way, having an opening to the sea through the hull at each end; the pump is only used when the vessel is at anchor, or, while getting under way. The draft is forced by one blowing engine and a fan, 3 feet in diameter. The boiler occupies a space 8½ by 8½ by 6 feet high, outside dimensions, and has one furnace. The grate surface is 55 and the heating surface, 1700 square feet. The propeller is of the Herreshoff type, four bladed, 5 feet 8 inches in diameter and 7 feet 2 inches pitch, and is made of aluminum bronze.

NOTE.—Such data as are now wanting to complete the tables will be published in future numbers of the journal.

STEAM TRIALS.

"VESUVIUS."—The results of the trials of this vessel have been looked for with more than ordinary interest, not only on account of the part she is intended to play in naval warfare, but also on account of the high speed stipulated in the contract—20 knots.

The following are her principal dimensions:

Length, ...	246 feet 3 inches.
Beam, ...	26 feet 6 inches.
Draught, mean, ...	9 feet 3 inches.
Displacement at above draught, .	805 tons.

She has twin-screw, vertical, triple-expansion engines, with cylinders 21½, 31, and two of 34 inches diameter, by 20 inches stroke, both engines being in the same compartment. The air pumps are worked off beams from the L. P. engines; the circulating pump (one for both engines) is of the centrifugal type. There are four boilers of the marine locomotive type, working at a pressure of 160 pounds; they are 9 feet in diameter and 19 feet 8 inches long, and have an aggregate grate surface of 200 square feet.

The first preliminary trial in Delaware river was made October 16, 1888, during which some slight defects were developed in the air pumps. On the second preliminary trial, November 26, the bell-crank of the valve-gear of one engine broke.

The first official trial, December 8, was made over a course 4.57 knots long. The first run over this course was with the tide, and the speed over the ground was at the rate of 21.47 knots per hour. The steam was 162 at the start and 144 at the finish, the mean revolutions being 264, the mean pressure in the cylinders being given as 59¾, 30½, 15, and 14⅜ respectively, and the collective I. H. P. 3,746.5. Revolutions of blowers, 600. On the return run the bed plate of one of the blower engines broke, and the speed for the run was not high enough to bring the average for the two runs up to 20 knots.

On the second official trial, December 27, she failed to make the required speed; average 19.57 knots. She was put over the

course again next morning, but this time one of the air pump beams broke.

The course for the last trial, January 11, 1889, was reduced to 2.5 knots, and was laid off in deep water near Delaware Breakwater. The first run, which was made with the tide, the vessel drawing 9 feet 6 inches, was at the rate of 22.947 knots, and the second at the rate of 20.346 knots, the mean being 21.646 or 1.646 knots more than the contract. The mean revolutions for these runs are given as 271.85, the boiler pressure 160, and the I. H. P., 4,295 without auxiliaries. As might be expected, the vibration in the cabin was quite marked during these runs.

On the afternoon of the day preceding the day of the last trial, she was subjected to trials at different speeds. The results, together with that for the official trial, are as follows:

Speed.	Revolutions.	I. H. P.
10.6	125.3	340
14.88	175.75	851
18.94	230.6	1,918
21.646	271.85	4,295

It may be of interest to note the force employed merely for running the engines during so short a trial. Besides the contractors' superintendent and the engineer in charge, there were eight machinists in the engine room, and two at the blowers, all of whom held certificates as chief engineers or first assistants in the merchant service. Besides these, there were eight men in the two fire rooms, making a total of twenty people, besides those engaged in taking data. The engine room was described as a mass of water and vapor.

"YORKTOWN."—This vessel has had several preliminary trials, and will soon be ready for the official trial.

"CHARLESTON."—Has not yet been tried, but is expected to be ready about the first of March.

"PETREL."—Will probably be ready for trial in March.

ENGLAND.

By far the most noteworthy trials which have taken place during the past six months are those of the battle ships "Victoria" and "Sanspareil," sister ships, of the following dimensions:

Length,	340 feet.
Beam,	70 feet.
Draught, mean load,	26 feet 9 inches.
Displacement,	10,470 tons.

The "Victoria" was built by the Thames Shipbuilding Company, and the "Sanspareil" by Sir William Armstrong, Mitchell & Co. The engines of both ships were designed and built by Humphrys, Tennant & Co., and are the most powerful triple-expansion engines which have yet been tried in any navy. They are vertical, twin-screw, having cylinders 43, 62, and 96 inches diameter, by 51 inches stroke, working with a boiler pressure of 135 pounds, and were designed to develop 7,500 I. H. P., with natural, and 12,000 with forced draft. The cylinders are supported on hollow steel columns, and the air pumps, as is usual in engines designed by this firm, are worked from the L. P. engines. Besides the main condensers, which have an aggregate cooling surface of 14,000 square feet, there is an auxiliary condenser in each engine room. Steam is supplied by eight four-furnace cylindrical boilers, placed in four water-tight compartments. The coal bunker capacity of each ship is 1,200 tons.

The following are the mean results of the trials, each of which was of four hours' duration, the ships in each case having been brought down to their designed load draught:

Name of Ship	VICTORIA.				SANSPAREIL.			
Natural or forced draft	Natural.*		Forced.		Natural.		Forced.	
Starboard or port engine.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.	Stbd.	Port.
Boiler pressure	131.	131.	133.	133.	133.6	133.6	130.	130.
Vacuum	27.8	26.8	27.4	27.0	27.5	27.5	26.	26.
Revolutions	87.3	88.5	103.26	100.37	86.86	87.37	100.76	100.7
I. H. P.	3,856.	4,182.	7,107.	7,137.	3,929.	4,110.	7,216.	7,266.
I. H. P., collective	8,038.		14,244.		8,039.		14,482.	
Speed by patent log	16.		17.25		16.		17.73	
Air pressure in fire rooms			1.97				1.90	

*In the English navy, this trial is made with what is called "open stokeholds", that is, the fire rooms are not closed air-tight, but the blowers are used to assist combustion.

The small gunboats of the "P" class ("Peacock," "Pigeon," "Pheasant," "Partridge," "Pigmy" and "Plover,") completed their trials recently, and the appended table gives the results for three of them. The others gave about the same results. They are composite vessels, of the following dimensions:

Length,	165 feet.
Beam,	30 feet.
Draught, mean load,	11 feet 4 inches.
Displacement,	755 tons.

All of them have single-screw, horizontal, triple-expansion engines, with cylinders 20, 30 and 45 inches diameter by 24 inches stroke, working at a boiler pressure of 140 pounds. The weight of machinery is 110 tons, and the coal supply 105.

The engines of the "Pheasant" and "Partridge" were built at Keyham dockyard; the others by the Naval Construction and Armament Company (Barrow).

The natural draft trials were of twelve hours' duration., these being the first vessels to come under the new Admiralty requirement for this trial; the forced draft trials were, in each case, of four hours' duration.

Name of vessel	"Peacock."		"Pigeon."		"Pheasant."	
Natural or forced draft.	N. D.	F. D.	N. D.	F. D.	N. D.	F. D.
Mean draught on trial	11′ 4″	11′ 4″	10′ 0″	9′ 8½″	11′ 3″	11′ 4″
Boiler pressure	139	142	142	143	134	138
Mean pressure, H. P. cyl	38.0	46.5	32.0	46.4	41.0	48.8
" " I. P.	18.3	22.4	15.0	22.1	17.7	24.2
" " L. P. "	8.7	14.3	10.51	13.4	9.3	15.0
Vacuum	25.5	22.0	25.0	22.1	25.0	24.2
Revolutions per minute	164.4	189.2	170	198.9	169.6	185.1
I. H. P., H. P.	238	336	233	352	265	344
" " I. P.	258	363	220	377	257	384
" " L. P.	275	522	338	513	302	536
" " total.	771*	1221†	791	1242	824‡	1264
Speed by pat. log	11.5	13.5	13.1	14.0	12.0	13.4
Coal per I. H. P.		2.3	1.95			

* On a previous trial, the I. H. P. was 690 for seven hours, but the trial had to be abandoned on account of priming in the boilers; fires were hauled, and the ship towed into port.

† Had a previous trial with forced draft.

‡ On her commissioning trial, with "open stokeholds," she developed 815 I. H. P., the revolutions being 160.5; steam, 130; vacuum, 24. By an Admiralty order, the engines are not permitted to be run so as to develop more power on this trial than hat developed on the contractor's trial with natural draft—but the fans may be used.

The "Nymphe" and "Daphne," twin-screw, composite sloops, have also, after a number of unsuccessful efforts, completed their trials. Their engines, which are horizontal triple-expansion, were built by the Greenock Foundry Co., a new aspirant for Admiralty work, and were designed for 1,400 I. H. P., with natural and 2,000 with forced draft. The following are their principal dimensions:

Length, ...	195 feet.
Beam	30 feet.
Draught, mean load, . .	12 feet 2 inches.
Displacement, tons, . .	1,140.

The cylinders are 18½, 29½ and 43½ inches diameter, by 24 inches stroke, working with a boiler pressure of 140 pounds. There are two three-furnace boilers.

The machinery is protected by a curved deck which is 9 inches above the water line over the boilers. The coal bunker capacity is 160 tons.

These little vessels carry an exceptionally heavy battery—eight 5-inch rifles, and eight machine guns—and are denominated "mistakes" by English naval officers.

On the "NYMPHE'S" first trial with "open stokeholds" the feed pumps gave so much trouble that the fires of both boilers had to be hauled; the vacuum was also very poor. After alterations in the feed pumps and air pump plungers, trials were had with the following results:

	Natural draft.	Forced draft
Mean draught, . .	12 feet 2 inches.	——
Boiler pressure, .	128	135
Revolutions, . .	149	169
Vacuum, . .	"indifferent"	21
I. H. P., . . .	1,440	2,076
Speed by patent log,	12.8	14.5
Coal per I. H. P., .	2	"Somewhat large".

THE "DAPHNE."—1,463 I. H. P. with natural, and 2,084 with forced draft, the revolutions in the last case being 170, and the speed 14.1 knots.

"MEDEA."—The trials of this vessel have been looked for with more than ordinary interest, from the fact that she is the first of the new 20-knot protected cruisers. She is built of steel and is of the following dimensions:

Length, ...	265 feet.
Beam, ...	41 feet.
Draught, mean load, . .	16 feet 6 inches.
Displacement, tons. .	2,800.

The only sail carried is fore-and-aft steadying sail on two masts.

The engines, by Humphrys, Tennant & Co., are twin-screw, vertical, triple-expansion, with cylinders 33½, 47 and 74 inches diameter, by 39 inches stroke, working with a boiler pressure of 155 pounds, designed to develop 5,500 I. H. P. with natural, and 9,000 with forced draft. The tops of the cylinders, which are 3 feet 6 inches above the water line, are protected by 5-inch steel coamings placed at an angle of about 25 degrees about the engine rooms. The boilers, of which there are four, are cylindrical double-ended, 12 feet diameter, with three furnaces at each end, The weight of the propelling machinery is 630 tons.

On her twelve hours' natural draft trial, the I. H. P. is reported as 6,100, the revolutions 130, and the speed about 18 knots. She has had several forced draft trials, two of them having to be discontinued on account of the breaking down of the fan engines and trouble with the feed pumps. On her last trial, the I. H. P. was 9,180, and the revolutions 140, but she failed to come within a knot of the speed of 20 knots for which she was designed, and now she is to have new screws.

The "Medusa," a sister ship to the "Medea," is nearing completion, as are also the "Marathon," "Melpomene," and "Magicienne," sheathed ships of the same class.

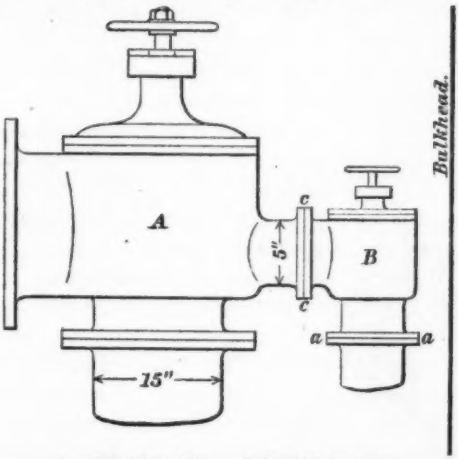
Since the trial of the "Medea" it has been decided to lay down an improved vessel of this class, to be 60 feet longer and to have 2 feet more beam.

CASUALTIES AND REPAIRS.

[It is the purpose of the Council to illustrate under this head the practical methods adopted on board ship for doing such work as would ordinarily have to be done in a shop, or ingenious methods which have been devised for doing such work as that described below. In sending in such items sketches should be made on plain white paper, and rolled, not folded.]

The following account of the method employed by Chief Engineer Baker, to examine the main injection valve of the "Lancaster", was communicated by Assistant Engineer Howard Gage.

Careful measurements were made of the position of the bolts securing the flanges *c c*, and a gasket of ⅛-inch cloth-lined rubber was made from these measurements, the hole in the centre of the gasket being left just large enough to admit a man's arm.



A, main injection valve. B, bilge injection valve.

Tackles were rigged to lift *B*, and blank flanges and backing were provided to block up the passage in *c c* should it be found necessary. Joints *c c* and *d d* were then broken, the valve lifted out, and, after a short struggle with the entering water, the rubber gasket secured to the face *c c*. On passing the arm through the hole in the gasket, the flow of water was stopped and the valve examined by touch.

B is about eight feet below the water line.

This examination was made because the main injection valve would not seat.

REVIEWS.

THE ROYAL NAVAL ENGINEERS' NOTE BOOK. By John Harvey, Fleet Engineer, R. N. Chatham: Gale and Polden.

This work, intended as a note book for engineers afloat, while partially filling a much needed want of Naval Engineers, is very disappointing to those who had formed their opinions from the press notices, and it is much to be regretted that the author did not increase its usefulness, which might have been done with little additional labor and at small increased cost of publication.

The information given on general engineering subjects of almost daily use is too meagre, and some important subjects are entirely omitted. The index, in not being alphabetically arranged, is almost valueless, and as the subjects are not alphabetically arranged it takes some time to find what you want. The space devoted to description of the ship and machinery is too small. The omission of tables showing valves to be opened or closed to pump out or fill a compartment is to be regretted. Altogether the book is a good one to build upon.

THE TECHNIC.—This annual of the Engineering Society of the University of Michigan for 1888 contains, amongst a mass of other readable matter, an illustrated description of the new Engineering Laboratory, which is under the direction of Prof. M. E. Cooley, formerly an Assistant Engineer U. S. Navy, and will be a fit monument to his zeal and industry. Prof. Cooley contributes an article on Dynamics of Reciprocating Engines, of which want of space forbids more than a mention.

THE "SANDFLY."

Among lessons to be learned from the naval manoeuvres, so far as they have gone, we must lay to heart the experience gained by a consideration of the eccentric action of the "Sandfly." Her misfortunes appear to be due to wrong construction, and not in any way to bad material, scamped work and inferior finish in sending her to sea. Whatever may be her qualities in smooth waters, the experience of her officers illustrates her value for such purposes as she is now put to. From their statements it seems that in light weather all the way from Portland, while the "Hercules" moved with the least possible motion, the "Sandfly" pitched and rolled so violently that the pendulum intended to register the heeling of the vessel proved useless. It registers only thirty

degrees on either side, and the "Sandfly" rolled nearly forty-five, the pendulum striking from side to side. * * * The ship's company to a man were sick, as men never were before. * * * On the bridge, which is as high as the funnels, the seas dashed up as they do against a cliff, lashing the salt water into the eyes of those on watch. * * * Forward of this bridge no man could go, for it was as freely under water as the bows of a cayak; and the four-inch breech-loader that stands there might as well be a champagne bottle, for any harm it could do an enemy while the vessel is under way. * * * The "Amphion" steamed easily her fifteen knots high out of water, while the "Sandfly" could make but six knots, and then rolled and pitched so that fighting of any kind was out of the question. * * * That she can neither make speed nor fight at sea is evident, and, what is worse, it is, in the mind of her officers, an open chance whether on any open sea distance she would reach her destination. Her short run has so battered her that now she is undergoing repairs. Her machinery has all the intricacy of clock-work, with much of its fragility. The slightest imperfection anywhere disables her, and no one on board was sanguine enough to think that she could float long unless she could keep her head to the sea. Her deck is so close to the water that one can sit and dip his toes in the water over the side, as was the case with the original monitors of the American Navy; but, unlike those vessels, the "Sandfly" has bulwarks that hold the water when it rushes over the side, and so destroy the ship's chance of freeing herself rapidly. * * * What the boat can do in smooth water is no doubt great, but as a war vessel, for general use, she has proved a complete failure. —*United Service Gazette, August* 4, 1888.

The correspondent of the *Army and Navy Gazette* has this to say about the behavior of the torpedo boats:

"No. 42 showed her keel from bow to midships, and then dipped till she almost stood on her nose. No. 61 had eighteen inches of water between decks, and had the ejector been used to pump it out, they would have had to ruin their boilers by using salt water."

The *Saturday Review* in an excellent article on "The Naval Manoeuvres," emphasises the opinion expressed by one of the *Times'* correspondents that the real cause of so many break-downs in the machinery of modern ships is to be found in the persistent endeavor of naval architects to cram three horses into a stable with stalls for two. The result is that engines and boilers cannot be made heavy enough—which is synonymous

with strong enough—to withstand the strains to which they are subject nowdays. They have no staying power. They may be excellent performers on the measured mile, but nowhere else, and never again. As the *Saturday* says, a ship that begins by making 14 knots, and keeps on making 14 knots, is better than one that begins at 18 and rapidly dwindles to 10, accomplishing even this latter speed with much puffing and groaning, and occasional pause to take breath. It will hardly be believed to what absurd lengths this saving of weight in machinery has been carried of late. We are ourselves acquainted with a ship, one of the belted cruisers, in which it was proposed by the Chief Engineer to fit a small piece of piping with the object of improving her feed arrangements; the length of the new fitting was, say, 3 feet, and the weight of it 20 pounds. Not long ago it would have been objected to on account of cost, but in this case it was refused on the score of the extra weight that would be involved, although its extreme usefulness was readily admitted by the authorities. This same vessel has been supplied with a bell which is big enough for a cathedral, and is hung from a bracket heavy enough to serve as a derrick for hoisting out her steam cutter. —*Army and Navy Gazette, Sept.* 1, 1888.

THE FOLLOWING particulars of the "belted cruiser," "Orlando," which are given in a Parliamentary return, the data having been furnished by the Admiralty under date of April 29, 1888, will give a very good idea of the distribution of weights in a modern ship.

The "Orlando" has twin-screw, horizontal triple-expansion engines, with cylinders 36, 52 and 78 inches diameter, by 42 inches stroke; cylindrical, double-ended boilers—grate surface 540 square feet, heating surface 16,055 square feet, steam pressure 130 pounds; condensing surface 12,000 square feet.

On her contractors' trial she developed 5,617 I. H. P. with natural draft, and 8,620 with forced draft. Before starting for Australia, where she is now stationed, she had a series of measured mile trials at different speeds, the mean of the four runs at full power giving 17.14 knots and 7,407 I. H. P., or only 86 per cent. of that maintained for four hours on the contractors' trial.*

* The First Lord of the Admiralty, in his statement explanatory of the navy estimates, 1888–89, says: "It is satisfactory to report that no less than 70 to 80 per cent. of the contract horse-power developed on the measured mile or six hours' contractors' trials, has been developed underthe severer conditions of 96 hours' continuous straining.

The ventilation of the fire rooms is so bad that en route to Australia the men could with difficulty be kept at work. The arrangement of coal bunkers is also faulty, necessitating the passing of the coal by hand from the forward bunkers. The extra work thus entailed, combined with the fact that the men were put on an allowance of a pint of water a day, owing to the fresh water distiller having broken down, almost led to a mutiny.

"Orlando"	As designed.	As completed.
	Feet. Inches.	*Feet. Inches.*
Length between perpendiculars	300 0	300 0
Breadth, extreme	56 0	56 0
Load draught, forward	19 6	21 0
aft	22 6	24 0
mean	21 0	22 6
Displacement.....tons	*5,040	*5,535
I. H. P.	†8,500	8,620
Full speed, knots per hour	18	18½
Height of armor above load water line	1 foot 6 inches.	Upper edge is just level with water line.
Depth of armor below water line	4 feet 0 inch,	5 feet 6 inches.
Coal at above load draught.....tons	†440	750
Capacity of bunkers	900	900
Armament	2 9.2-inch (18 ton) 10 6-inch (4½ ton). 6 6-pdr., R. F. G. 10 1-inch Nordenfeldt; boat and field guns; torpedoes	2 9.2-inch (22 ton). 10 6-inch (5 ton). 6 6-pdr., R. F. G. 10 3-pdr., R. F. G.; boat, field and small machine guns; torpedoes.
Complement of officers and men	350	455
Weights:		
Water and tanks		
Provisions and tare		
Officers' stores and slops		
Officers, men and effects		
Masts, spars and rigging	365	394
Anchors and cables		
Boats		
Warrant officers' stores		
Guns, carriages and slides, powder, projectiles, torpedoes, electrical stores, &c.	350	430
Machinery, propelling	800	795
Engines for electric lighting, capstan, steering, torpedoes, &c.	‡15	‖34
Engineers' stores	45	37
Coals	440	750
Conning tower	80	80
Director tower	.....	5
Hull and armor	2,945	¶3,010
Total.....tons	5,040	5,535

*The displacements are for the ship as laid off. (The British Navy Estimates give it as 5,600 and the mean draught as 22 feet 8 inches.—Ed.)

†The design upon which tenders were invited specified compound engines of 7,500 i. h p, with 500 tons of coal on board, altered in accepting tender to triple-expansion engines with 440 tons of coal.

‡Electric light not included.

‖Including electric light.

¶Includes weight due to adoption of central power hoist arrangements, electric lighting, bridge, increased accommodation for largely augmented crew, davits, skid beams and other fittings for additional boats, and increased thickness of armor at ends of belt.

EXAMINATIONS.

U. S. Navy Regulation Circular No. 52.

NAVY DEPARTMENT,
WASHINGTON, *December* 26, 1888.

U. S. Navy Regulation Circular No. 16, dated November 17, 1874, is hereby rescinded, and the following substituted therefor:

Before an Assistant Engineer can be promoted to the grade of Passed Assistant Engineer in the Navy, he must have been examined by a board of officers of the Medical Corps and found physically qualified, and have established, to the satisfaction of a board of officers of the Engineer Corps, his mental, moral and professional qualifications to perform efficiently all the duties, both at sea and on shore, of the grade to which he is to be promoted.

A candidate for promotion to the grade of Passed Assistant Engineer must have served at least three years at sea as an Assistant Engineer on board a naval steamer.

The testimonial letters concerning him, required by the Regulations from all the commanding and senior engineer officers under whom he may have served, will be placed before the examining board by the Department.

The candidate will be examined on the following subjects:

1. Testimonials.

2. Details of construction of the various marine boilers commonly used, and of their attachments. Causes of deterioration of boilers. Means of preserving boilers. The different methods of applying forced draft. Causes of priming, and the means used to remedy and prevent it. The different methods used to increase the circulation of water in boilers, and their advantages and disadvantages.

3. Principles, peculiarities and details of construction of the different kinds of modern steam expansion valves, valve-gears, condensers, air and circulating pumps, feed pumps and their connections. Derangements in their operation, the causes thereof, and the means used to prevent and correct them. Details of construction of the various steam pumps in use, and the advantages and disadvantages of each.

[The examination on the subjects included in paragraphs 2 and 3 will be on modern practice as far as possible, so that the candidate may have an opportunity of showing that he is familiar with the recent changes in modern naval and marine engineering.]

4. Practical building and repairing steam machinery, including the details of molding and casting. The candidate will be required to trace the progress of a part or piece of the machinery

of a naval vessel (the subject to be designated by the board) from the raw material, through the various stages of making and fitting, to the finished piece in the mechanism, and to show his knowledge of the various processes and tools employed during this work in the pattern and boiler shops, foundry, smithery, machine and fitting shops, and in placing the piece in its position in the vessel.

5. Theory, laws and construction of the various kinds of screw-propellers. Theory of the steam engine, the calculations connected therewith, and the interpretation of indicator diagrams.

6. Strength of materials and the influence of form in connection therewith. Tests of materials. Description of the different kinds of dynamos in use.

7. A working drawing showing the necessary dimensions of some piece of the engines or boilers of a naval vessel, from data given by the board, the component parts thereof to be proportioned to the work and strength required of them. The drawing to be made in the presence of, and to be graded by, the board, according to the correctness of the design and the proficiency in mechanical drawing.

8. The candidate will be required to write, in the presence of the board, an original essay, not less than five nor more than ten pages of foolscap in length, on some engineering subject which the candidate has investigated, made a study of, or experimented in. In grading the essay, especial weight will be given to the importance of the subject and to the manner of treating it, but the style of the language will also be marked.

9. French, or any other foreign language with which the candidate is familiar.

VALUATION OF SUBJECTS.

No. 1,	100
No. 2,	120
No. 3,	170
No. 4,	150
No. 5,	90
No. 6,	100
No. 7,	100
No. 8,	90
No. 9,	80
Total,	1,000
Lowest satisfactory,	700

WILLIAM C. WHITNEY,
Secretary of the Navy.

U. S. Navy Regulation Circular No. 53.

NAVY DEPARTMENT,
WASHINGTON, D. C., *December* 26, 1888.

U. S. Navy Regulation Circular No. 46, dated December 7, 1885, is hereby rescinded, and the following substituted therefor:

Before a Passed Assistant Engineer can be promoted to the grade of Chief Engineer in the Navy, he must have been examined by a board of officers of the Medical Corps and found physically qualified, and have established, to the satisfaction of a board of officers of the Engineer Corps, his mental, moral and professional qualifications to perform efficiently all the duties, both at sea and on shore, of the grade to which he is to be promoted.

A candidate for promotion to the grade of Chief Engineer must have served at least two years at sea as a Passed Assistant Engineer on board a naval steamer.

Interrogatories will be addressed by the board to commanding and senior engineer officers under whom the candidate may have served for answers as to his mental, moral and professional fitness for promotion.

The candidate will be examined on the following subjects:

1. Calculations for the design of engines, boilers, propellers and auxiliary machinery of a naval vessel, upon data to be given by the board, and the proportioning of such principal parts of the engines as the board may require. The candidate to lay out the general plans of the machinery to suit a given space, and to make a general working drawing of one of the boilers. He will be allowed the use of such tables and formulæ as are necessary to enable him to make his design.

2. Practical building and repairing machinery. Making and fitting the various parts of marine engines, auxiliary machinery, boilers and screw-propellers. Precautions to be observed during the inspection of the work in its various stages. Erecting, securing and aligning the machinery on board wooden, iron or steel vessels. The candidate to show a thorough familiarity with, and knowledge of, the various processes employed in the different shops; also, with the preparation of the machinery for the preliminary and others trials. Derangements in, and breakdowns of, naval machinery, and the means adopted to remedy and repair them.

3. Advantages and disadvantages of the different kinds of marine engines (simple and multiple expansion). General theory and practice of iron and steel ship-building.

4. Deterioration of boilers, and causes and remedies thereof. Practical metallurgy. Tests of materials.

5. The candidate will be required to write, in the presence of the board, an original essay, not less than seven pages of foolscap in length, on engineering matters in general.

[This essay is intended to give the candidate an opportunity to express his views on the progress of engineering science, and on the changes and improvements made in the practice of marine engineering. The essay will be marked with reference to the value and correctness of the subject-matter, and to the style of the composition.]

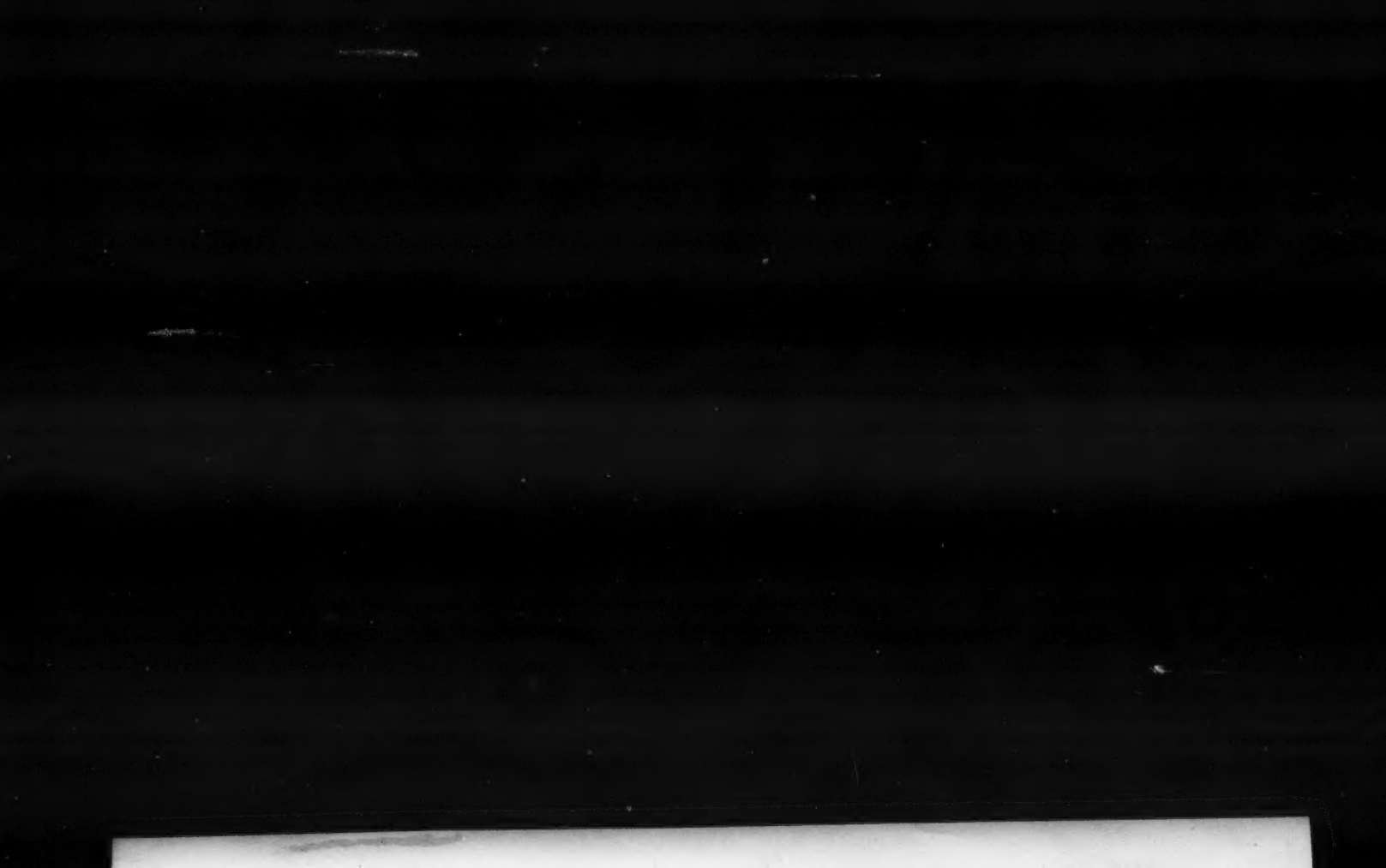
VALUATION OF SUBJECTS.

No. 1,	270
No. 2,	220
No. 3,	170
No. 4,	170
No. 5,	170
Total,	1,000
Lowest satisfactory	700

WILLIAM C. WHITNEY, *Secretary of the Navy.*

OBITUARY.

The Engineer Corps of the Navy has met with a loss in the death of Chief Engineer Edwin Wells, who died on board the "Omaha," at Shanghai, China, January 8, 1889.

He was one of the older and well known members of the Corps, and an ornament to the service socially, as well as professionally. His name always awakened among his old shipmates the memories of his genial and kindly wit, and during his long period of service he has not left a single enemy, but all his acquaintances were his friends.

He was born at St. Mary's, Chester County, Pa., September 15, 1837; entered the service as Third Assistant Engineer, July 3, 1861; promoted to Second Assistant, December 18, 1862; to First Assistant, January 30, 1865, and to Chief Engineer, May 31, 1880. He served during the war from July, 1861, to February, 1865, on the steamers "Rhode Island" and "San Jacinto", of the Blockading Squadron; from 1865 to 1867 on the "Winooski"; 1867 to 1872 on the "Michigan", "Contoocook", "Quinnebaug", and "Wasp"; September, 1872, to November, 1873, at Naval Station, New London; to January, 1876, as member of Naval Examining Board, Washington; to 1879 on the "Tallapoosa"; 1880 to 1883 on the "Marion"; 1884 to 1886 on duty connected with the new cruisers; on the "Omaha" from June 22, 1886, until his death, January 8, 1889.